| L Number | Hits | Search Text | DB | Time stamp |
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| 1 | 1 | ("3417660").PN. | USPAT | 2003/05/09 11:22 |
| 3 | 14 | 3417660.URPN. | USPAT | 2003/05/09 11:20 |
| 4 | 41 | damping adj3 particles | USPAT | 2003/05/09 11:47 |
| 5 | 0 | 6543590.URPN. | USPAT | 2003/05/09 11:40 |
| 6 | 20 | ("1294467" \| "2155052" \| "2417347" \| "2732040" \| "2869700" \| "3456782" \| "3637051" \| "3899100" \| "3938625" \| "4289419" \| "4350233" \| "4557466" \| "4706788" \| "4858738" \| "4934661" \| "5020644" \| "5306100" \| "5486078" \| "6401628" \| "0121414").PN. | USPAT | 2003/05/09 11:40 |
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| 16 | 5625 | vibration same particles | USPAT | 2003/05/09 11:47 |
| 17 | 22099 | vibration$1 and damping | USPAT | 2003/05/09 11:47 |
| 18 | 2194 | (vibration$1 and damping) and particles | USPAT | 2003/05/09 11:47 |
| 19 | 405 | ((vibration$1 and damping) and particles) and (damping same particles) | USPAT | 2003/05/09 11:48 |
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| 23 | 20 | ("1894276" \| "2862686" \| "3586460" \| "4019301" \| "4057250" \| "4182512" \| "4453887" \| "4566231" \| "5016602" \| "5020644" \| "5098098" \| "5180163" \| "5195930" \| "5197707" \| "5327733" \| "5345177" \| "5400296" \| "5454562" \| "5772540" \| "5775049").PN. | USPAT | 2003/05/09 12:03 |
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| 35 | 7 | 5020644.URPN. | USPAT | 2003/05/09 12:12 |
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| 38 | 5 | 1995620.URPN. | USPAT | 2003/05/09 12:13 |
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| 42 | 3 | 5182888.URPN. | USPAT | 2003/05/09 12:16 |

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| 43 | 5 | ("4499694" \| "4593502" \| "4633628" \| "4713917" \| "4899323").PN. | USPAT | 2003/05/09 12:17 |
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| 45 | 4 | ("3938852" \| "4050665" \| "4483426" \| "4566231").PN. | USPAT | 2003/05/09 12:18 |

A. H. KING.

Improvement in Car Springs.

No. 123,999. Patented Feb. 27, 1872.

*Fig. 1.*

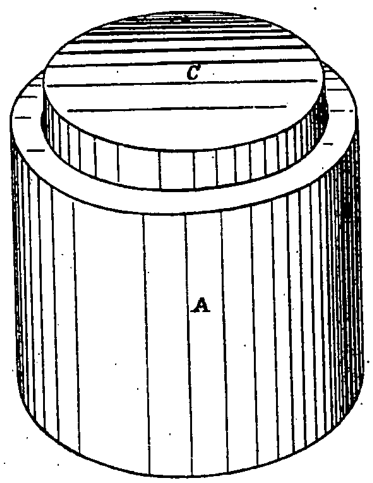



*Fig. 2.*

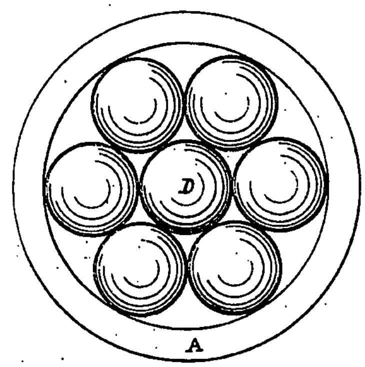



*Fig. 3.*

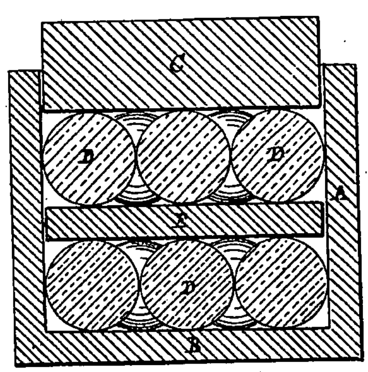
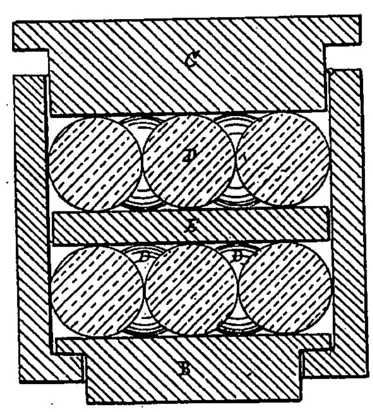



*Fig. 4.*

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George F. Stenz

Inventor.
Albert H. King
By Wm. C. Wood
Attorney

# UNITED STATES PATENT OFFICE.

ALBERT H. KING, OF RAHWAY, NEW JERSEY.

## IMPROVEMENT IN CAR-SPRINGS.

Specification forming part of Letters Patent No. 123,999, dated February 27, 1872.

*To all whom it may concern:*

Be it known that I, ALBERT H. KING, of Rahway, in the county of Union and State of New Jersey, have invented a certain new and useful Car-Spring.

My invention relates to that general class of springs which depend for elastic force upon vulcanized rubber or caoutchouc; and consists in combining within a chamber and between suitable base and crown plates a requisite number of solid spheroidal masses of elastic vulcanized rubber; and I do hereby declare that the following specification, taken in connection with the drawing furnished and forming a part of the same, is a clear, true, and exact description of a car-spring embodying my invention.

Referring to the drawing, Figure 1 represents one of my car-springs in perspective. Fig. 2 represents the same in top views with crown-plate removed. Fig. 3 represents the same in vertical section. Fig. 4 represents in vertical section a similar spring provided with a movable base-plate.

A denotes the spring-chamber. In this instance it is represented as cylindrical. Equally good results may be attained if it be made square, rectangular, or in any form between the latter and the cylindrical. B denotes the base-plate. It may be incapable of independent movement, and form a part of the chamber, as shown in Fig. 3; or it may be made separately therefrom, and capable of movement within the cylinder, as shown in Fig. 4. C denotes the crown-plate or follower. It is fitted to enter freely the interior of the chamber, and may or not be provided with a projecting lip or flange. D denotes the spheroidal elastic masses of vulcanized rubber. They are molded to the requisite form in a manner well known. As exhibited in the drawing, the chamber A contains fourteen of these spheroids, two series, seven in each, with a separating plate, E, interposed between the series. When a light spring is required good results may be attained by the use of a single spheroidal mass within a suitable chamber, the inside vertical walls of which are adjacent to the horizontal periphery of the spheroid, and interposed directly between the base and crown-plate. For a spring in which greater range of action is requisite, several series of single spheroidal masses may be employed within a longer chamber, with or without the interposition of the separating plates. For a spring which is not to be subjected to great strain in carrying heavy weights a single series of two or more solid spheroidal masses, arranged side by side, may be employed within a chamber the vertical walls of which should be closely adjacent to the exterior peripheries of the spheroids lying next to the walls. Should a greater range of action be required and suited for carrying the same weight, two or more series, each consisting of several solid spheroidal masses, may be employed without the interposition of separating-plates between the series. A spring composed of two or more series of several solid spheroids each, with interposed separating-plates between the series, will meet the general and varied requirements of ordinary rail service. There exists in a solid spheroidal mass of elastic vulcanized rubber or caoutchouc a constant tendency to maintain its true form. If subjected to a lateral as well as vertical pressure its expansive tendencies are always in all directions toward its entire periphery from the center. When confined within certain limits by adjacent walls of a chamber, or by contact with other masses of a similar form and character, and interposed between movable plates to which pressure is applied, each spheroidal mass contributes of its force in a comparatively-equal degree, and as the pressure is increased the elastic force of the several masses is combined to resist it. That portion of each spheroid above and below its horizontal axial line constitutes what may be properly considered a combined mass of elliptical springs, radiating from each end of its vertical axis toward the periphery at the horizontal axial line, and therefore approximate results will be attained if the spheroidal masses be cut in two at their horizontal axial lines. Practice and experiment has demonstrated that a spring composed of rubber or caoutchouc in a solid spheroidal mass, and arranged substantially as herein shown and described by me, for sustaining a given weight and exercising a desired degree of elastic force while loaded, can be produced by using from twenty-five to fifty per cent. less rubber or caoutchouc than is requisite for practically constructing a spring in any of the methods heretofore practiced and known to me, and which would be capable of meeting equal require-

ments. I am aware that chambers have been filled with irregular masses of rubber or caoutchouc and other substances of a non-elastic character, but such springs have never proven to be of practical value. After long service, should the spheroids become flattened to any observable extent, they may be slightly turned in position, and other portions thereof presented for the contact of the adjacent surfaces. By judicious management of this character such springs will prove to be capable of long and continued service with a uniform degree of efficiency.

Having thus described my invention, I claim as new and desire to secure by Letters Patent—

1. A mass of solid vulcanized elastic rubber or caoutchouc, spheroidal in form, substantially as described, in combination with a chamber for receiving the same, the interior vertical walls of which are adjacent to the periphery of the mass when in position at the horizontal line of its axis, a base-plate and a crown-plate, one or both of which are capable of independent vertical movement within the chamber, as and for the purposes specified.

2. The combination of two or more solid masses of elastic vulcanized India rubber or caoutchouc, spheroidal in form, substantially as described, with a chamber, a base-plate, and a crown-plate, one or both of which are capable of an independent vertical movement within the chamber, as and for the purposes specified.

3. A car-spring, composed of two or more solid spheroidal masses of elastic vulcanized India rubber or caoutchouc, substantially as described, arranged in two or more horizontal series, one above the other, with separating plates between the series, confined within a chamber, and interposed between a base and a crown-plate, as and for the purposes specified.

ALBERT H. KING.

Witnesses:
WM. C. WOOD,
PHIL. F. LARNER.

# United States Patent [19]

**Fukahori et al.**

[11] **Patent Number:** **4,899,323**

[45] **Date of Patent:** **Feb. 6, 1990**


[54] **ANTI-SEISMIC DEVICE**


[75] Inventors: **Yoshihide Fukahori**, Hachioji; **Hiromu Kojima**, Higashimurayama; **Akihiko Ogino**; **Shigenobu Suzuki**, both of Kodaira; **Toshikazu Yoshizawa**, Hachioji, all of Japan

[73] Assignee: **Bridgestone Corporation**, Tokyo, Japan



[21] Appl. No.: **337,045**

[22] Filed: **Apr. 12, 1989**

**Related U.S. Application Data**

[63] Continuation-in-part of Ser. No. 306,263, Feb. 1, 1989, abandoned, which is a continuation of Ser. No. 78,621, Jul. 28, 1987, abandoned.

[30] **Foreign Application Priority Data**

| | | | |
|---|---|---|---|
| Aug. 4, 1986 | [JP] | Japan | 61-183196 |
| Sep. 16, 1986 | [JP] | Japan | 61-217689 |
| Oct. 2, 1986 | [JP] | Japan | 61-234897 |
| May 2, 1988 | [JP] | Japan | 63-109604 |

[51] **Int. Cl.$^4$** .......... **E04B 1/98**

[52] **U.S. Cl.** .......... **367/176**; 52/167 DF; 248/560; 248/638

[58] **Field of Search** .......... 367/176; 52/167; 248/560, 638; 181/146, 151, 166



[56] **References Cited**

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*Primary Examiner*—Thomas H. Tarcza
*Assistant Examiner*—Ian J. Lobo
*Attorney, Agent, or Firm*—Kanesaka and Takeuchi



[57] **ABSTRACT**

An anti-seismic device which comprises anti-seismic rubber bearings and dampers arranged in parallel, the anti-seismic rubber bearing being formed by laminating a plurality of rigid hard plates and soft boards having a viscoelastic property one over another, the damper being composed mainly of a viscoelastic material having the physical properties (i) and (ii) defined below.

(i) the hysteresis ratio ($h_{50}$) is greater than 0.3 at 50% tensile deformation at 25° C.

(ii) the storage modulus (E) measured dynamically at a frequency of 5 Hz, a strain of 0.01%, and a temperature of 25° C. is in the range of $1 \leqq E \leqq 2\times10^4$ (kg/cm$^2$).

**20 Claims, 18 Drawing Sheets**

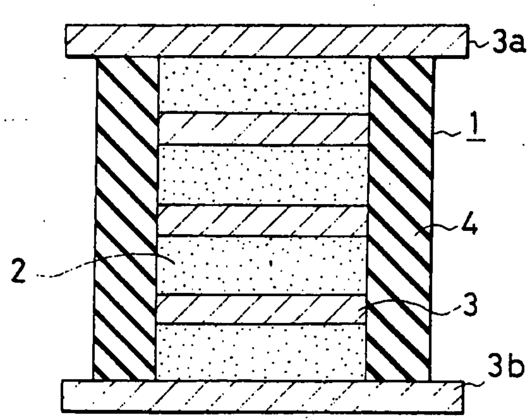



FIG. 1

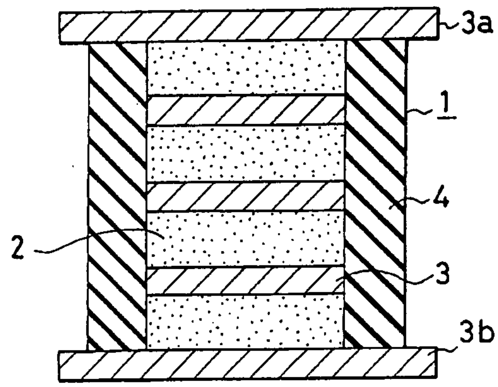



FIG. 2

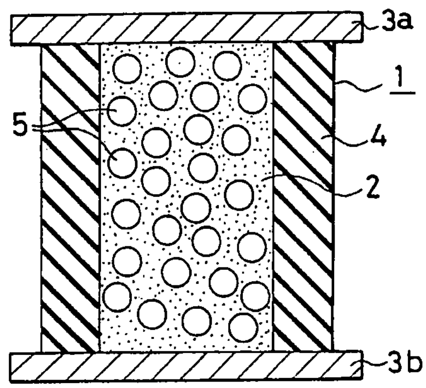



FIG. 3

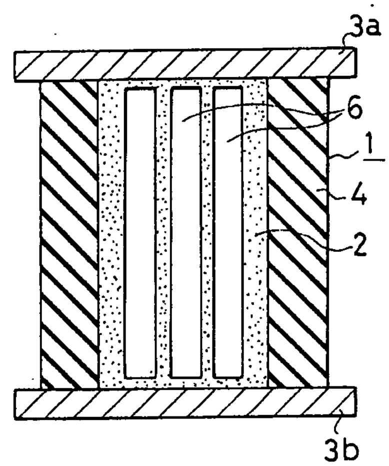


FIG.4

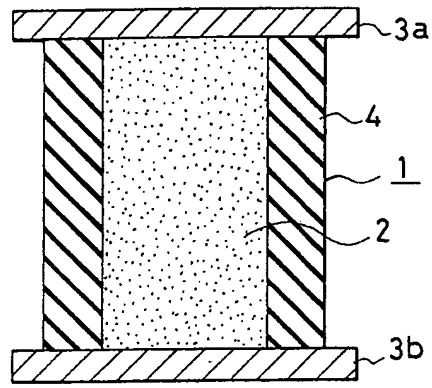



FIG.5

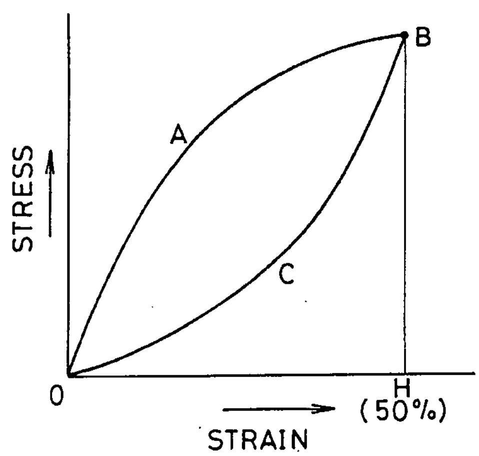


FIG.6
(a)

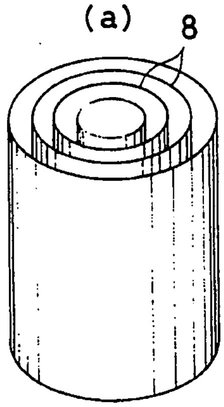



FIG.6
(b)

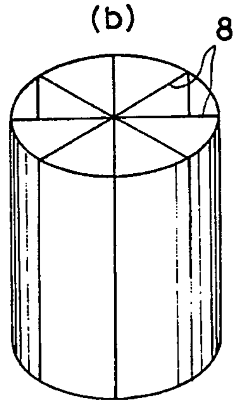



FIG.6
(c)

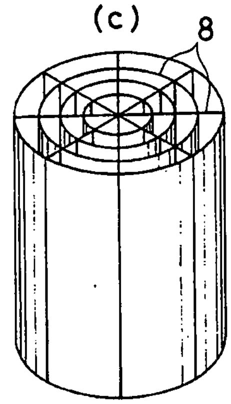



FIG.6
(d)

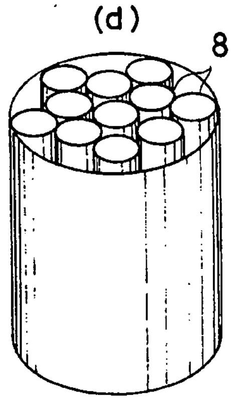



FIG.6
(e)

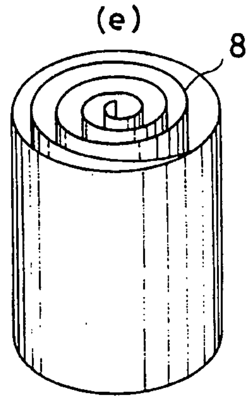


FIG.7

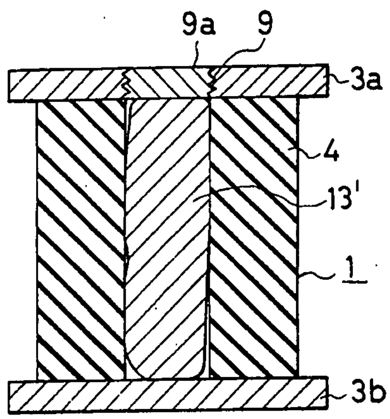



FIG.8

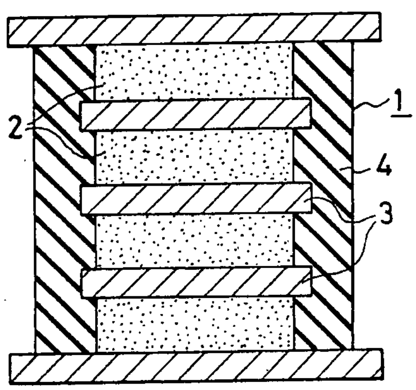



FIG.9

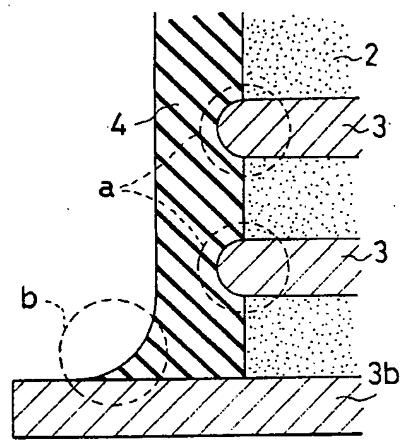


FIG.10

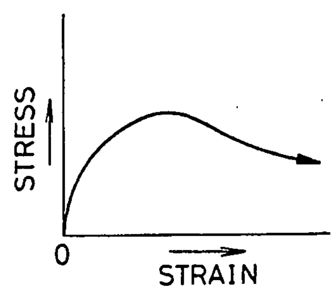
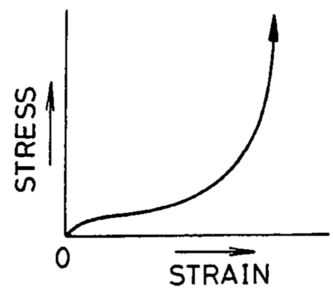



FIG.11

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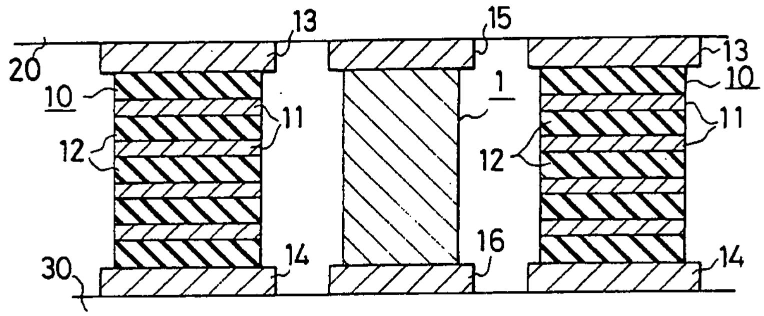



FIG.13

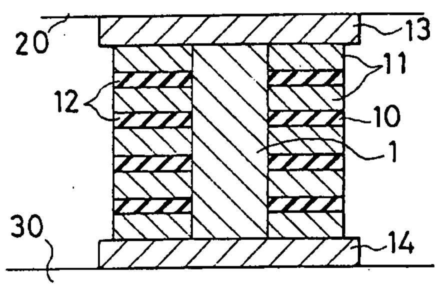


FIG.14

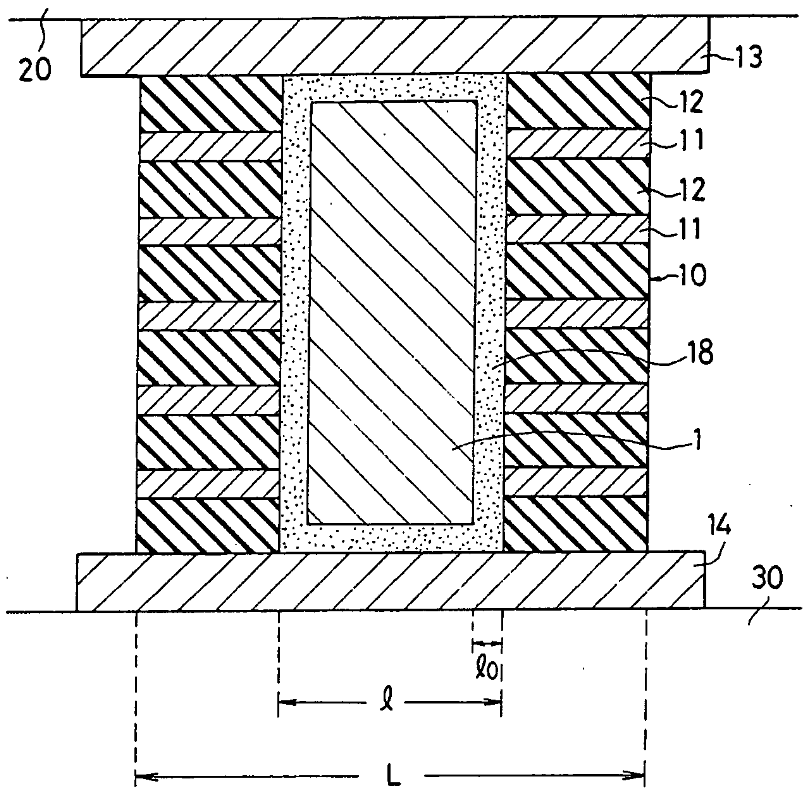


FIG. 15

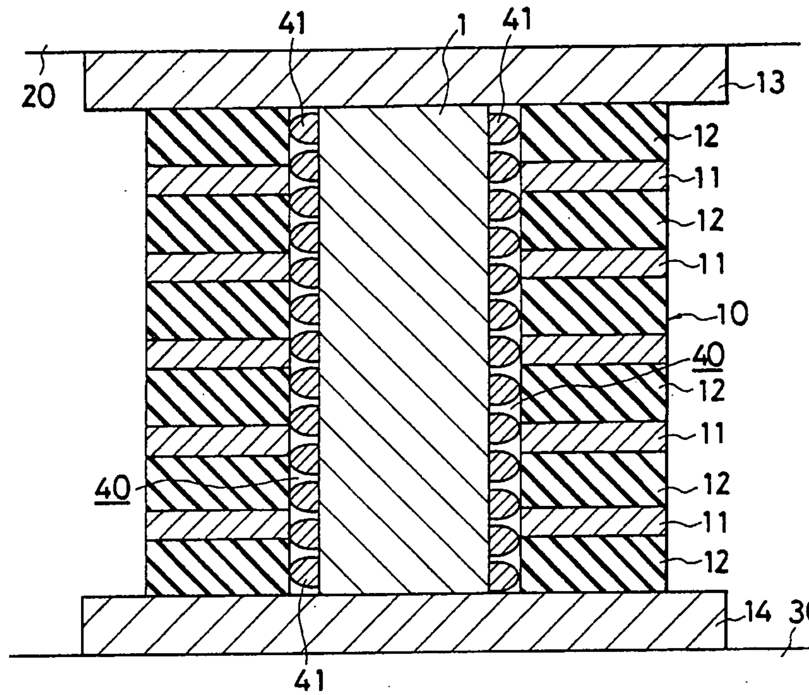


FIG.16

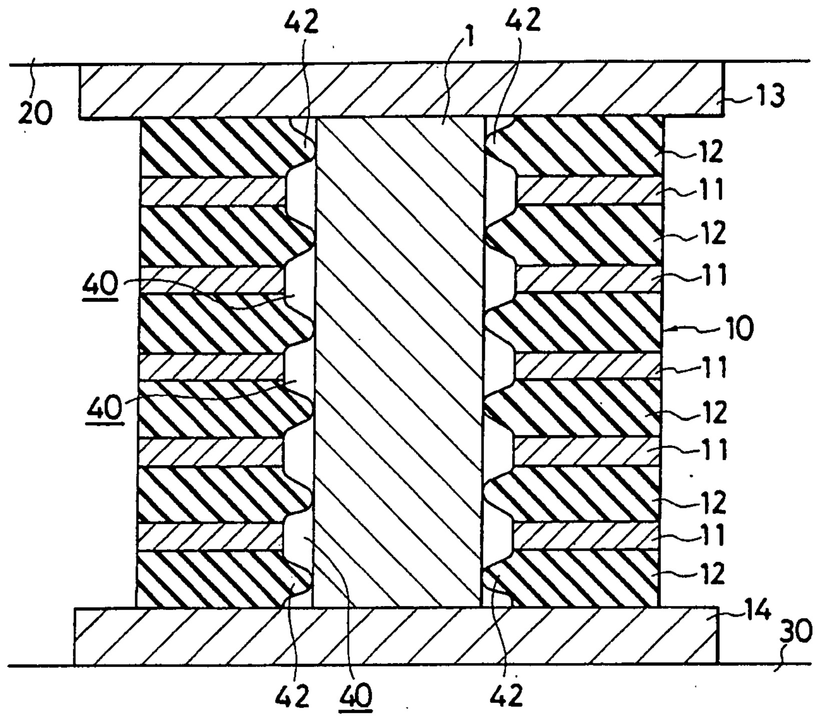


FIG.17

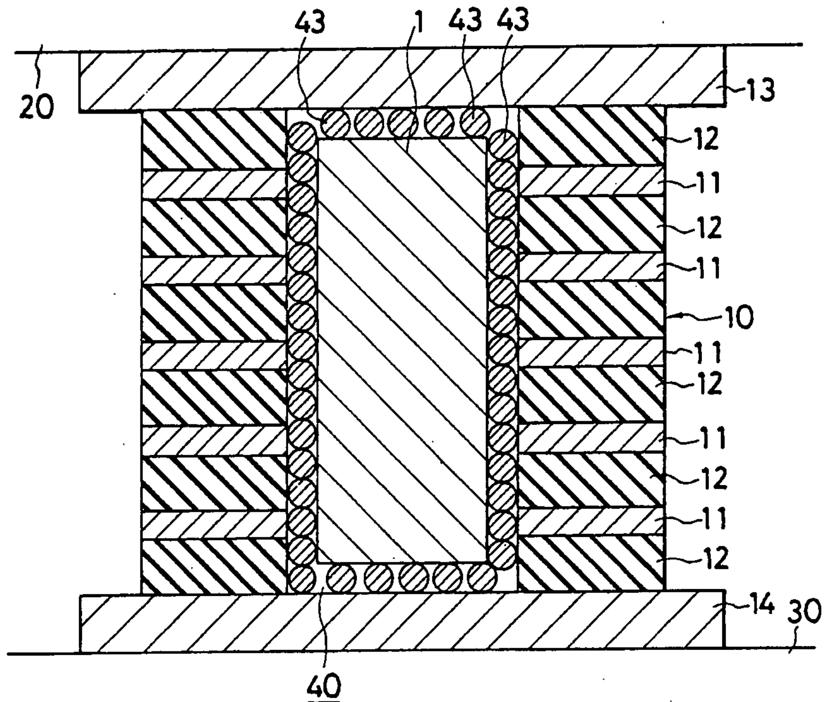
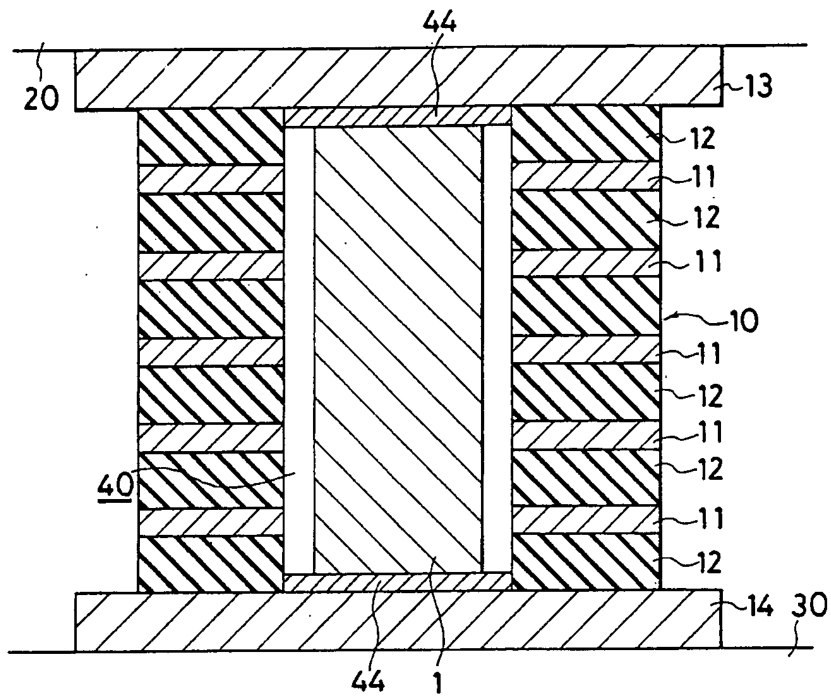


FIG.18



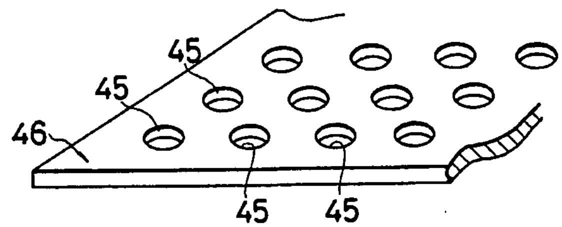



FIG.19

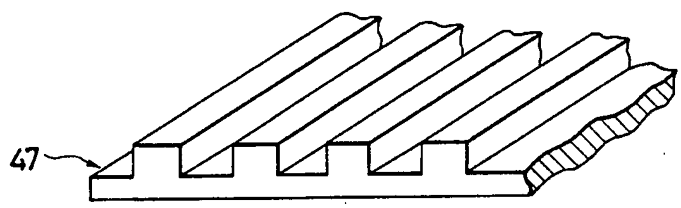



FIG.20

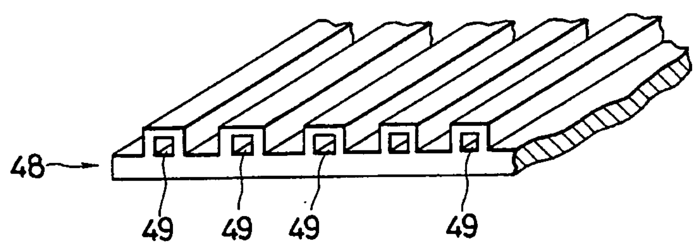
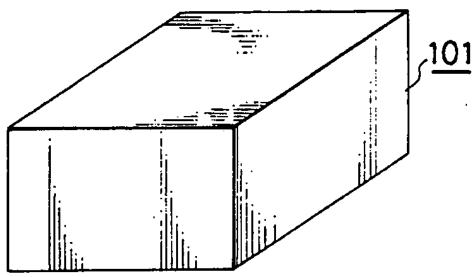



FIG.21

FIG.22

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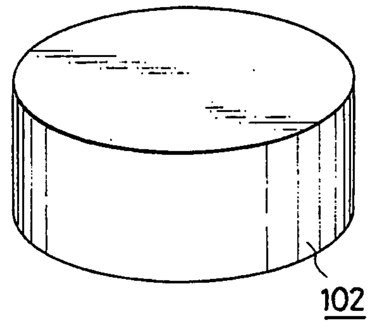



FIG.24

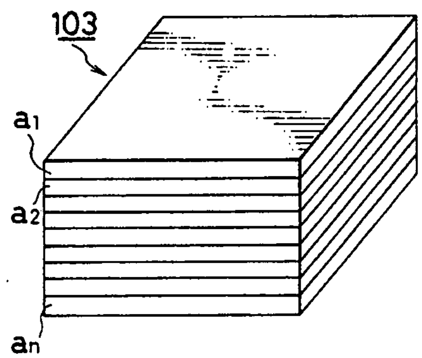



FIG.25

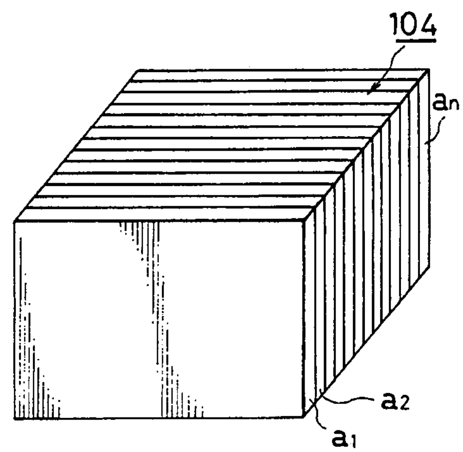


FIG.26

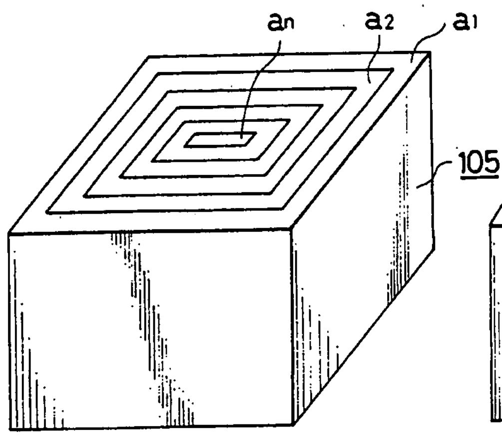



FIG.27

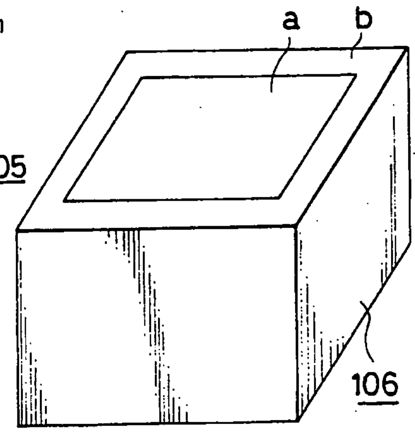



FIG.28

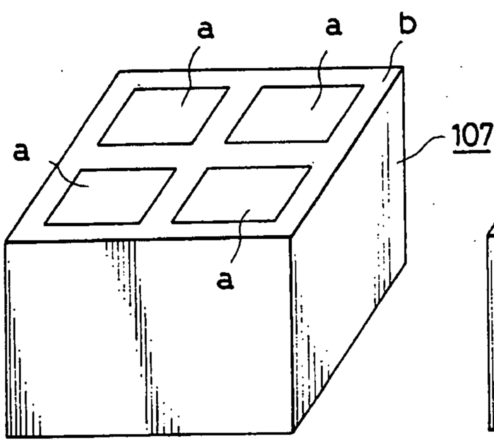



FIG.29

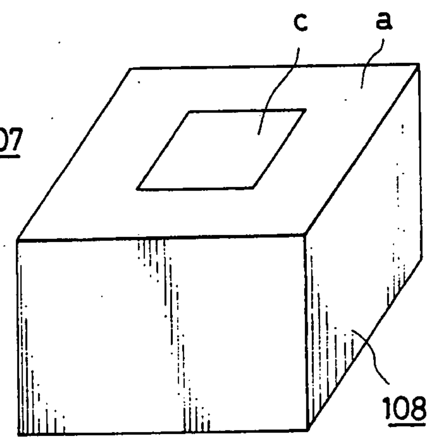


FIG.30

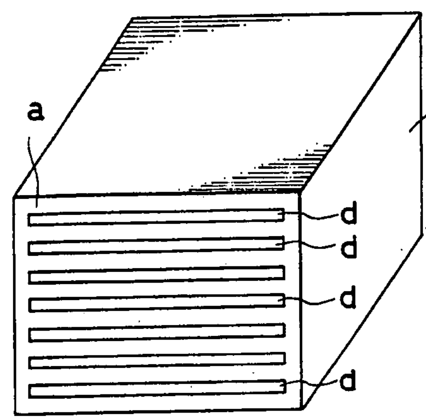



FIG.31

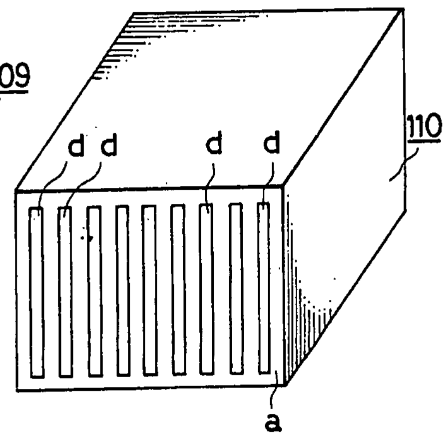



FIG.32

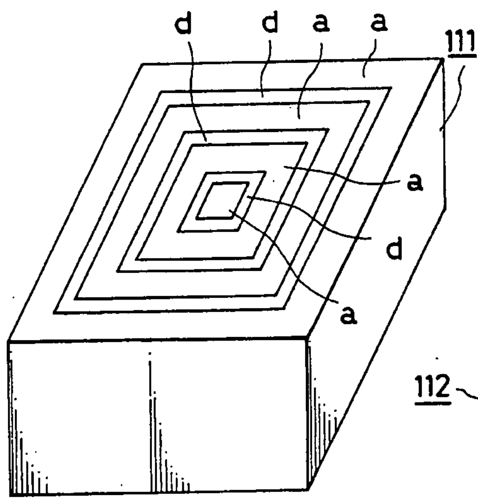



FIG.33

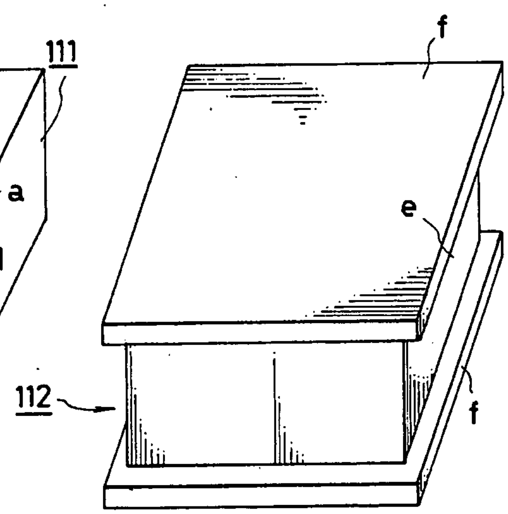


FIG.34

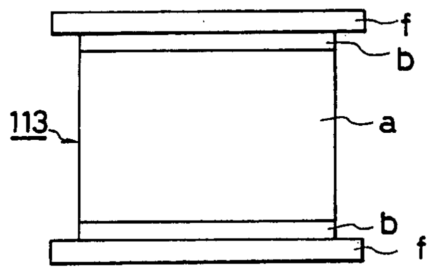



FIG.35

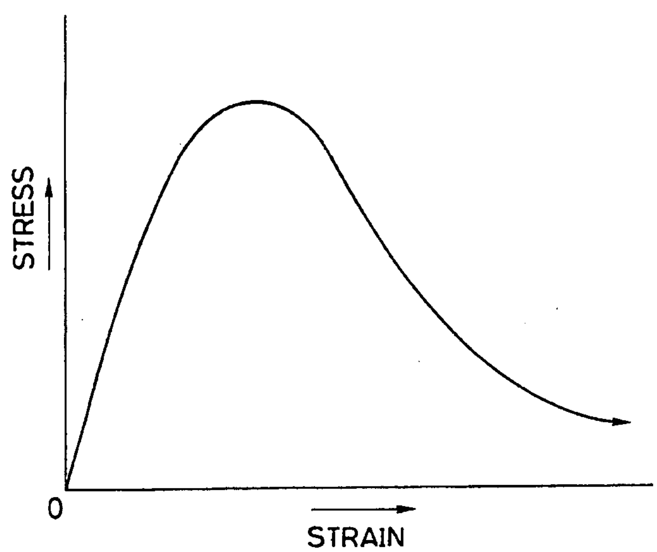
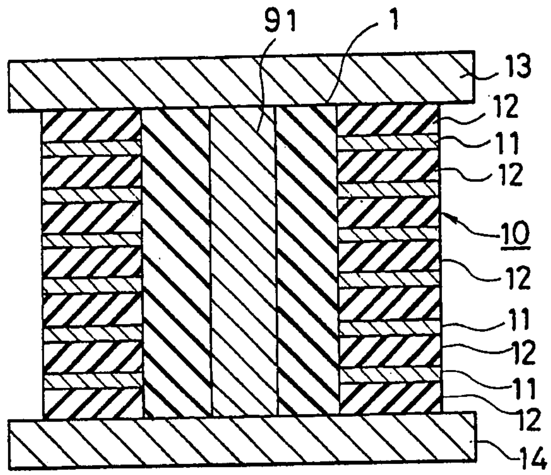


FIG.36


91
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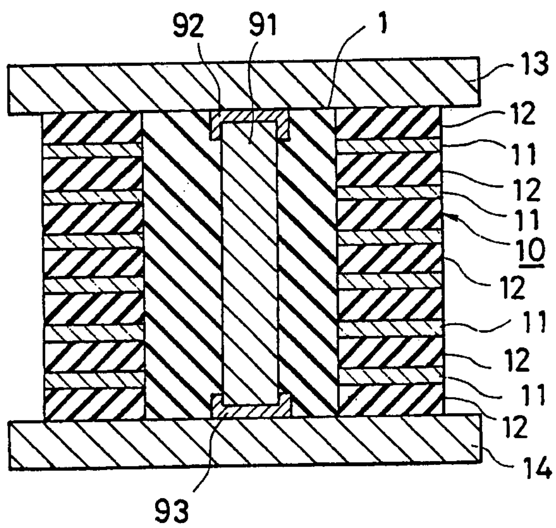
14

FIG.37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93

FIG.38

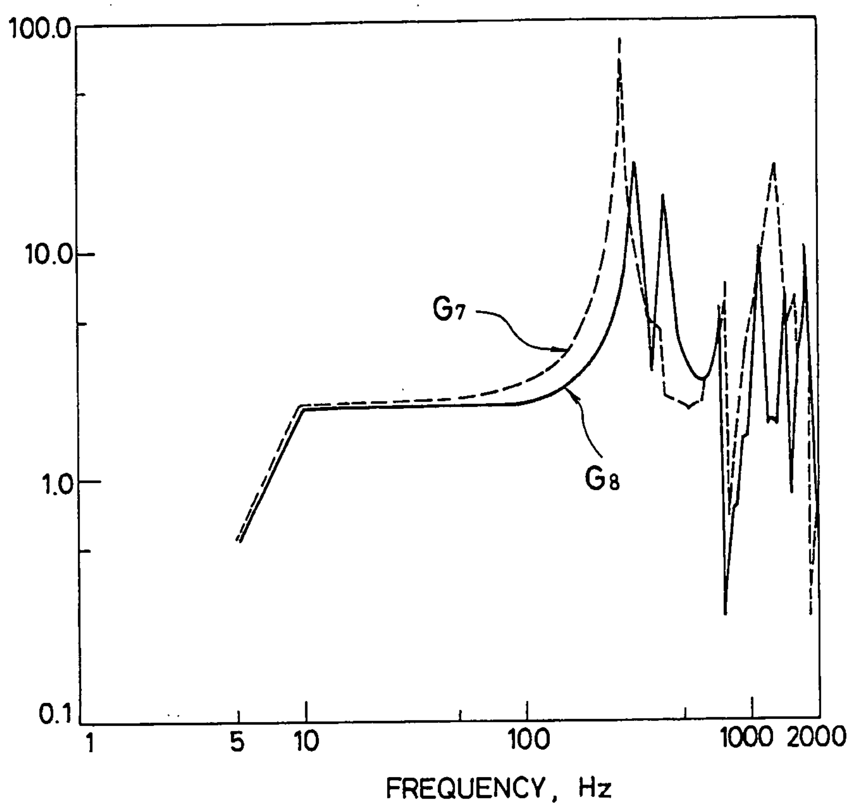
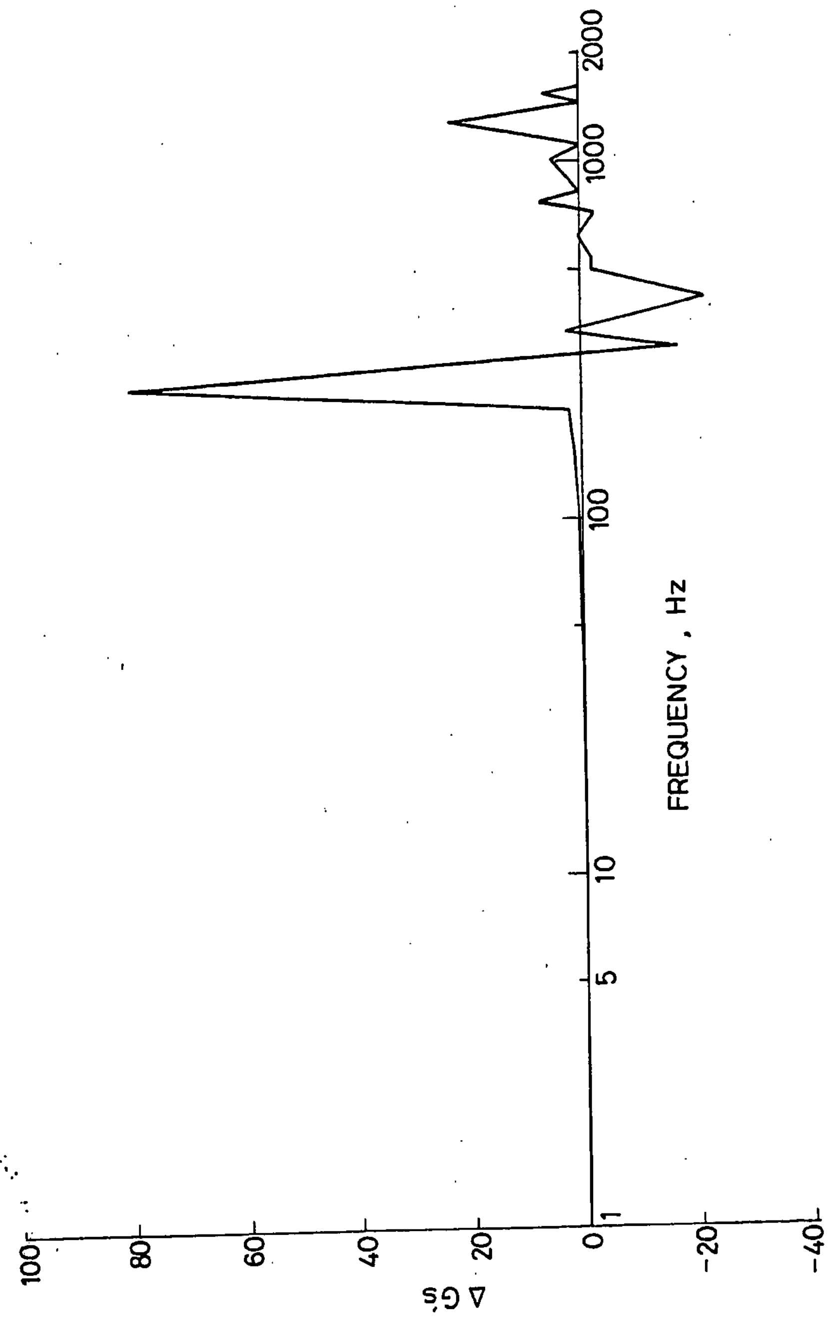


FIG. 39



## ANTI-SEISMIC DEVICE

### CROSS-REFERENCE TO RELATED APPLICATIONS

This is a CIP application of U.S. Ser. No. 306,263 filed on Feb. 1, 1989, now abandoned, which in turn is a continuation of application Ser. No. 078,621 filed on July 28, 1987 abandoned.

### FIELD OF THE INVENTION AND RELATED ART STATEMENT

The present invention relates to an improvement of the anti-seismic device which consists of dampers and anti-seismic rubber bearings arranged in parallel, thereby producing both the anti-seismic effect and the damping effect, said dampers absorbing vibrational energy or earthquake motion (abbreviated as vibration hereinafter) applied to machines and structures at the time of earthquake.

Dampers have long been known as a means to reduce vibration applied to machines and structures by an earthquake. Their ability to absorb vibrational energy is derived from the material of which they are constructed. They fall into two main groups; those which utilize the plastic effect of a metal such as lead, and those which utilize the viscous effect of oil.

Much attention is now focused on a body (anti-seismic rubber bearing) formed by laminating alternately a plurality of steel plates and rubber boards. It is designed to protect an object it supports from vibration at the time of earthquake.

Such an anti-seismic rubber bearing is flexible (or has a low shear modulus) in its lateral direction. Because of this property, anti-seismic rubber bearings interposed between a rigid structure such as a concrete building and the foundation thereof shift the natural frequency of the structure from the seismic frequency, thereby greatly reducing the acceleration applied to the structure by an earthquake.

A feature of the anti-seismic rubber bearing of this type is its ability to undergo elastic deformation which permits the restoration to its original shape after an earthquake. However, the anti-seismic rubber bearing itself has an extremely limited energy absorbing ability (damping effect) so that the subsidence of a building resulting from its creeping is minimized. For this reason, the conventional anti-seismic rubber bearing is made of rubber having a low hysteresis loss.

In the case of an anti-seismic device composed of the above-mentioned low damping anti-seismic rubber bearings alone, the structure installed on them continues to shake horizontally for a long time even after an earthquake is over. This horizontal shaking, if excessive, may cause damage to the anti-seismic rubber bearings themselves as well as the building and utility such as water pipes, gas pipes, and wirings.

The conventional way to diminish the horizontal shaking in as short a time as possible was to combine an anti-seismic rubber bearing with a plastic damper of soft metal or the like which undergoes plastic deformation as soon as it receives a seismic force. For example, this damper is formed by filling a void in the anti-seismic rubber bearing with lead. The thus formed device produces both the anti-seismic effect and the damping effect.

A disadvantage of the conventional damper utilizing the plastic effect is that it produces the damping effect



very little when the deformation is small, in which case the deformation is elastic deformation.

A disadvantage of the conventional damper utilizing the viscous effect of oil is that it has to be large in size if it is to produce a considerable damping effect. An additional disadvantage is that the handling of oil needs care, the fabrication is difficult, and the complex maintenance work is required for use over a long period of time.

The anti-seismic device having a conventional built-in plastic damper absorbs more seismic energy than that without a plastic damper; however, it in turn has a shortcoming of resonating in the high-frequency region because the plastic damper has a high elastic modulus.

The lead-filled anti-seismic rubber bearing has a disadvantage that when it greatly deforms at the time of large earthquake the hard plates such as steel plates damage the lead and the damaged lead in turn damages the soft boards such as rubber boards, which leads to the entire breakage of the anti-seismic rubber bearing. The damaged lead tends to break easily when it undergoes great deformation repeatedly.

There is disclosed a vibration damping stiffener in U.S. Pat. No. 4,566,231. This produces almost no damping effect at the vibration frequencies of earthquakes.

### OBJECT AND SUMMARY OF THE INVENTION

It is an object of the present invention to provide an anti-seismic device which produces both the anti-seismic effect and the damping effect. The anti-seismic device is provided with a viscous damper which is characterized by:

(1) its constituent material which exhibits the maximum viscosity effect (damping effect),

(2) its structure which permits the constituent material to fully exhibit its viscosity effect (damping effect),

(3) ease of molding and fabrication,

(4) easy handling, and

(5) low cost.

The anti-seismic device of the invention comprises anti-seismic rubber bearings and dampers arranged in parallel, the anti-seismic rubber bearing being formed by laminating a plurality of rigid hard plates and viscoelastic soft board on top of the other, the damper being composed mainly of a viscoelastic material having the physical properties (i) and (ii) defined below.

(i) the hysteresis ratio ($h_{50}$) is greater than 0.3 at 50% tensile deformation at 25° C.

(ii) the storage modulus (E) measured dynamically at a frequency of 5 Hz, a strain of 0.01%, and a temperature of 25° C. is in the range of $1 \leqq E \leqq 2 \times 10^4$ (kg/cm²).

In order to eliminate the disadvantages of the conventional viscous damper, the present inventors carried out extensive studies on the ideal viscous damper having the above-mentioned merits (1) to (5). As the result, it was found that the satisfactory damping effect is produced when the damper is made of a viscoelastic material which has the hysteresis ratio, Mooney viscosity, and storage modulus in a certain range. It was also found that when the dampers of this type are combined with anti-seismic rubber bearings in parallel, the resulting anti-seismic device supports a building stably for a long period of time and reduces the vibration transmitted to the building it supports. These findings led to the present invention.

## BRIEF DESCRIPTION OF THE DRAWINGS

FIGS. 1 to 4 are longitudinal sectional views of the dampers pertaining to the present invention.

FIG. 5 is a graph showing a stress-strain curve of a material.

FIGS. 6(*a*) to 6(*e*) are perspective views showing the partitioning members.

FIG. 7 is a longitudinal sectional view illustrating how to make the damper pertaining to the present invention.

FIG. 8 is a longitudinal sectional view of a damper in one embodiment of the invention.

FIG. 9 is a partly enlarged longitudinal sectional view showing the flange and its vicinity in another embodiment of the invention.

FIG. 10 is a graph showing a stress-strain curve of a viscoelastic material.

FIG. 11 is a graph showing a stress-strain curve of a skeleton.

FIG. 12 is a longitudinal sectional view of the anti-seismic device pertaining to an embodiment of the invention.

FIG. 13 is a longitudinal sectional view of the anti-seismic device pertaining to another embodiment of the invention.

FIG. 14 is a longitudinal sectional view of the anti-seismic device pertaining to further another embodiment of the invention.

FIGS. 15 to 18 are longitudinal sectional views showing the anti-seismic device of the invention.

FIGS. 19 to 21 are perspective views showing the material to be arranged around the damper.

FIGS. 22 to 34 are perspective views showing the preferred examples of the damper.

FIG. 35 is a stress-strain curve of unvulcanized rubber.

FIGS. 36 and 37 are sectional views showing another embodiment of the anti-seismic device.

FIGS. 38 and 39 are diagrams showing the characteristic properties of the conventional damping stiffener.

## DESCRIPTION OF THE PREFERRED EMBODIMENTS

In what follows, a detailed description will be made of the present invention.

The damper as a constituent member of the anti-seismic device of the present invention is made mainly of a viscoelastic material having the physical properties as defined in sections (i) and (ii) below.

(i) The hysteresis ratio ($h_{50}$) at 50% tensile deformation at 25° C. is greater than 0.3, preferably greater than 0.35, more preferably greater than 0.4. The hysteresis ratio ($h_{50}$) is the ratio of are to area OABCO to area OBAHO in the stress-strain curve (at a pull speed of 200 mm/min) shown in FIG. 5.

(ii) The storage modulus of elasticity (E) dynamically measured at a frequency of 5 Hz, a strain of 0.01%, and a temperature of 25° C. is in the range of $1 \leqq E \leqq 2 \times 10^4$ (kg/cm$^2$), preferably greater than $1 \times 10^4$ (kg/cm$^2$), more preferably greater than $5 \times 10^3$ (kg/cm$^2$), and most desirably greater than 1 and smaller than $2 \times 10^3$ (kg/cm$^2$).

The viscoelastic material should have an elongation (at tensile break) greater than 1%, preferably greater than 5%, more preferably greater than 10%, and most desirably greater than 20%.

The viscoelastic material for the damper in the present invention includes unvulcanized rubber, vulcanized rubber, and synthetic resins and plastic materials having the above-mentioned characteristic properties.

According to the present invention, the viscoelastic material should preferably be unvulcanized rubber, vulcanized rubber, or a similar material having the above-mentioned hysteresis ratio and elastic modulus. Their examples include common rubber such as ethylene-propylene rubber (EPR, EPDM), nitrile rubber (NBR), butyl rubber, halogenated butyl rubber, chloroprene rubber (CR), natural rubber (NR), isoprene rubber (IR), styrene-butadiene rubber (SBR), butadiene rubber (BR), acrylic rubber, ethylene-vinyl acetate rubber (EVA), and polyurethane rubber; special rubber such as silicone rubber, fluororubber, ethylene-acrylic rubber, polyester elastomer, epichlorohydrin rubber, and chlorinated polyethylene; and thermoplastic elastomers. If the viscoelastic material is unvulcanized rubber, it is desirable that the Mooney viscosity $ML_{1+4}$ at 100° C. be higher than 10.

These rubber materials may be used alone or in combination with one another. In addition, they may be incorporated with additives such as filler, tackifier, slip agent, antioxidant, plasticizer, softening agent, low-molecular weight polymer, and oil which are commonly used for rubber processing to impart desired hardness, loss characteristics, and durability according to the object of use. Where the rubber materials are required to maintain the desired performance over a long period of time, they should be stabilized by adding a proper antioxidant, polymerization inhibitor, anti-scorching agent, etc. and/or by modifying the polymer itself by hydrogenation, etc.

Where it is necessary to bond the viscoelastic material to other constituent material, the bonding may be advantageously accomplished by utilizing the stickiness of the viscoelastic material. To ensure the bonding by stickiness, a network structure may be formed in the interface by chemical bonding or physical bonding.

The viscoelastic material in the present invention may be selected from the following materials in addition to the above-mentioned unvulcanized rubber and vulcanized rubber: thermoplastic resins such as polystyrene, polyethylene, polypropylene, ABS, polyvinyl chloride, polymethyl methacrylate, polycarbonate, polyacetal, nylon, polyether chloride, polytetrafluoroethylene, polyfluorochloroethylene, polyfluoroethylene-propylene, acetyl cellulose, ethyl cellulose, polyvinylidene, vinyl butyral, and polypropylene oxide; and rubber-modified products thereof: thermosetting resins such as epoxy resin and unsaturated polyester, and rubber-modified products thereof. These plastics may be incorporated with the following filler, plasticizer, softening agent, tackifier, oligomer, slip agent, antioxidant, and low-molecular weight polymer oil according to need. These plastics may be used alone or in combination with one another.

(a) Filler: Flaky inorganic filler such as clay, diatomaceous earth, carbon black, silica, talc, barium sulfate, calcium carbonate, magnesium carbonate, metal oxide, mica, graphite, and aluminum hydroxide; granular or powdery filler such as metal powder, wood chips, glass powder, ceramics powder, and polymer powder or granules; and natural and artificial short fibers and long fibers (such as straw, wool, glass fiber, metal fiber, and polymer fiber), which are used for rubbers and resins.

The amount of the filler should preferably be 30–250 parts by weight for 100 parts by weight of rubber.

The short fibers include glass short fiber, plastics short fibers, and natural short fibers. They also include the following special reinforcing short fibers. The short fibers should preferably be chemically bonded to the rubber just like the molecule of vulcanizable rubber is grafted with short fibers of thermoplastic polymer having the

$$-\overset{\overset{\displaystyle O}{\|}}{C}-NH-$$

group through a precondensate of phenol-formaldehyde resin. The thermoplastic polymer for short fibers should be one which has the —CONH— group in the polymer molecule. Examples of this polymer include nylons such as nylon-6, nylon-610, nylon-12, nylon-611, and nylon 612; polyurea such as polyheptamethylene urea and polyundecamethylene urea; and polyurethane, which have a melting point of 190°–235° C., preferably 190°–225° C., and more preferably 200°–220° C. They should be added in an amount of 30–250 parts by weight. The short fibers should have an average diameter of 0.05–0.8 μm and a circular cross section. The short fiber should have a minimum fiber length greater than 1 μm. In addition, the short fiber should be composed of molecules oriented in the axial direction.

(b) Softening agent: Aromatic, naphthenic, and paraffinic softening agents for rubbers and resins.

The preferred amount of the softening agent is 5–150 parts by weight for 100 parts by weight of rubber.

(c) Plasticizer: Ester-type plasticizer such as phthalic ester, mixed phthalic ester, ester of dibasic aliphatic acid, glycol ester, ester of fatty acid, phosphoric ester, and stearic ester; epoxy-type plasticizer; and other plasticizers for plastics. Phthalate-, adipate-, sebacate-, phosphate-, polyester-, and polyester-type plasticizers for NBR.

The preferred amount of the plasticizer is 5–150 parts by weight for 100 parts by weight of rubber.

(d) Tackifier: Coumarone resin, coumarone-indene resin, phenol-terpene resin, petroleum hydrocarbon, and rosin derivative.

The preferred amount of the tackifier is 1–50 parts by weight for 100 parts by weight of rubber.

(e) Oligomer: Fluorine-containing oligomer, polybutene, xylene resin, chlorinated rubber, polyethylene wax, petroleum resin, rosin ester rubber, polyalkylene glycol diacrylate, liquid rubber (such as polybutadiene, styrene-butadiene rubber, butadiene-acrylonitrile rubber, and polychloroprene), silicone oligomer, and poly-α-olefins.

The preferred amount of the oligomer is 5–100 parts by weight for 100 parts by weight of rubber.

(f) Slip agent: Hydrocarbon-type slip agent such as paraffin and wax; fatty acid-type slip agent such as higher fatty acid and hydroxy fatty acid; fatty acid amide-type slip agent such as alkylene bisfatty acid amide; ester-type slip agent such as lower alcohol ester of fatty acid, polyhydric alcohol ester of fatty acid, and polyglycol ester of fatty acid; alcohol-type slip agent such as aliphatic alcohol, polyhydric alcohol, polyglycol, and polyglycerol; and metal soap; and mixtures thereof.

The preferred amount of the slip agent is 1–50 parts by weight for 100 parts by weight of rubber.

In the present invention, the viscoelastic material may be a natural product such as bitumen and clay. However, unvulcanized rubber having the above-mentioned characteristic properties is most preferable when all factors are taken into account.

In the meantime, rubber in an unvulcanized state is poor in restoring property and flows slowly with time, losing its shape after a long period of time. Therefore, in the case where the viscoelastic material used for the damper is a soft material such as unvulcanized rubber, it is necessary to cover it with vulcanized rubber or other proper material so that the flow of the unvulcanized rubber is prevented. This surface covering also permits the damper to undergo great deformation.

The vulcanized rubber used for covering (referred to as "covering rubber" hereinafter) may be produced from any of the above-mentioned vulcanized rubber. In other words, the unvulcanized rubber as the internal viscoelastic material of the damper may have a formulation identical with or similar to that of the vulcanized rubber for the covering layer. Needless to say, they may be entirely different from each other.

For the damper of the present invention to be durable for a long period of time, the covering rubber should be made of one having good weather resistance. Examples of weather-resistant rubber include butyl rubber, acrylic rubber, polyurethane, silicone rubber, fluororubber, polysulfide rubber, ethylene-propylene rubber (ERP and EPDM), Hypalon, chlorinated polyethylene, ethylene-vinyl acetate rubber, epichlorohydrin rubber, and chloroprene rubber. Of these, butyl rubber, polyurethane, ethylene-propylene rubber, Hypalon, chlorinated polyethylene, ethylene-vinyl acetate rubber, and chloroprene are especially effective.

These rubber materials may be used individually or in combination with one another. For the improvement of their elongation and other physical properties, they may be blended with commercial rubber such as natural rubber, isoprene rubber, styrene-butadiene rubber, butadiene rubber, and nitrile rubber. In addition, these rubber materials may be incorporated with additives such as filler, antioxidant, plasticizer, softener, and oil which are commonly used for rubber processing. A preferred rubber compound is composed of 100 parts by weight of rubber, 10–40 parts by weight of cyclopentadiene or dicyclopentadiene resin, and 5–20 parts by weight of rosin. This rubber compound is greatly improved in fracture properties and adhesion to metals.

The following experiments were conducted in order to make clear the properties of the rubber materials of the invention.

Rubber materials Nos. 1 to 13 and resin materials Nos. 14 to 16 shown in the following tables were examined for physical properties, namely, hysteresis ratio ($h_{50}$), storage modulus (E), and elongation at tensile break.

The compositions of the materials are indicated in the column "composition" in the tables. Rubber materials Nos. 1 to 4 are not cured, rubber materials Nos. 5 and 6 are cured slightly, and rubber materials Nos. 10 to 13 (which are comparative examples) are cured fully. (The slight curing is accomplished by using the crosslinker in a less amount, say 10 to 60%, than usual.) Resin materials Nos. 14 to 16 are not cured.

In the tables, PVC stands for polyvinyl chloride, PVAc stands for vinyl chloride-vinyl acetate copolymer, BBP stands for butyl benzyl phthalate, and DCPD stands for dicyclopentadiene resin.

The hysteresis ratio $h_{50}$ and the storage modulus E are measured in the same manner as described in the specification of the present application.

The test piece used for measuring $h_{50}$ is a ring in shape having an outside diameter of 25 mm and an inside diameter of 21 mm. The test piece used for measuring E is 50 mm long, 5 mm wide, and 2 mm thick. The test piece used for measuring elongation at tensile break is a dumbbell in shape conforming to JIS K6301. It was pulled at a rate of 200 mm/min at 25° C.

above-mentioned properties, said hard plates and said soft layers being laminated one over another, with the external surface of said soft layers covered with vulcanized rubber.

(2) One which is composed of a viscoelastic material having the above-mentioned properties and a solid substance embedded in said viscoelastic material, with the external surface of the soft body covered with vulcanized rubber.

(3) One which is composed merely of a viscoelastic

| | 1 | | 2 | | 3 | | 4 | |
|---|---|---|---|---|---|---|---|---|
| Composition (parts by weight) | NR | 100 | SBR | 100 | EPDM | 100 | CR | 100 |
| | Carbon black | 20 | Carbon black | 20 | Carbon black | 60 | Carbon black | 20 |
| | Zinc oxide | 5 | Zinc oxide | 5 | Zinc oxide | 5 | Zinc oxide | 5 |
| | Antioxidant | 3 | Antioxidant | 3 | Antioxidant | 3 | Antioxidant | 3 |
| | | | | | DCPD | 20 | | |
| | | | | | Rosin | 10 | | |
| Curing conditions | not cured | | not cured | | not cured | | not cured | |
| Properties | | | | | | | | |
| $h_{50}$ | 0.60 | | 0.71 | | 0.65 | | 0.63 | |
| $E(kg/cm^2)$ | 31 | | 69 | | 150 | | 87 | |
| Elongation at break (%) | 100 and up | | 100 and up | | 100 and up | | 100 and up | |
| | 5 | | 6 | | 7 | | 8 | |
| Composition (parts by weight) | NR | 50 | EPDM | 100 | EPDM | 70 | NR | 67 |
| | BR | 50 | Graphite | 150 | NR | 30 | BR | 33 |
| | Carbon black | 85 | Aromatic oil | 20 | Carbon black | 60 | Carbon black | 80 |
| | DCPD | 60 | Sulfur | 0.45 | Zinc oxide | 5 | Zinc oxide | 5 |
| | Phenolic resin | 30 | | | Antioxidant | 3 | Antioxidant | 3 |
| | Sulfur | 0.45 | | | DCPD | 20 | DCPD | 60 |
| | | | | | Rosin | 10 | Sulfur | 1.5 |
| | | | | | Sulfur | 1.5 | | |
| Curing conditions | slightly cured | | slightly cured | | fully cured | | fully cured | |
| Properties | | | | | | | | |
| $h_{50}$ | 0.55 | | 0.48 | | 0.48 | | 0.43 | |
| $E(kg/cm^2)$ | 1020 | | 250 | | 220 | | 330 | |
| Elongation at break (%) | 100 and up | | 100 and up | | 100 and up | | 100 and up | |
| | 9 | | 10 | | 11 | | 12 | |
| Composition (parts by weight) | BR | 67 | NR | 70 | NR | 30 | SBR | 30 |
| | NR | 33 | BR | 30 | BR | 70 | Carbon black | 20 |
| | Carbon black | 70 | Carbon black | 15 | Carbon black | 70 | Zinc oxide | 5 |
| | Zinc oxide | 5 | Zinc oxide | 5 | Zinc oxide | 5 | Antioxidant | 3 |
| | Antioxidant | 3 | Antioxidant | 3 | Antioxidant | 3 | Sulfur | 1.5 |
| | DCPD | 30 | DCPD | 5 | Sulfur | 1.5 | | |
| | Aromatic oil | 15 | Sulfur | 1.5 | | | | |
| | Sulfur | 1.5 | | | | | | |
| Curing conditions | fully cured | | fully cured | | fully cured | | fully cured | |
| Properties | | | | | | | | |
| $h_{50}$ | 0.38 | | 0.07 | | 0.19 | | 0.25 | |
| $E(kg/cm^2)$ | 150 | | 26 | | 190 | | 23 | |
| Elongation at break (%) | 100 and up | | 100 and up | | 100 and up | | 100 and up | |
| | 13 | | 14 | | 15 | | 16 | |
| Composition (parts by weight) | EPDM | 100 | PVC | 85 | Low-density PE | 100 | PMMA | 100 |
| | Carbon black | 20 | PVAc | 15 | | | BBP | 70 |
| | Zinc oxide | 5 | BBP | 100 | | | Carbon black | 10 |
| | Antioxidant | 3 | Glass powder | 100 | | | | |
| | Sulfur | 1.5 | | | | | | |
| Curing conditions | fully cured | | not cured | | not cured | | not cured | |
| Properties | | | | | | | | |
| $h_{50}$ | 0.24 | | 0.87 | | 0.76 | | 0.81 | |
| $E(kg/cm^2)$ | 35 | | 8500 | | 9800 | | 9100 | |
| Elongation at break (%) | 100 and up | | 100 and up | | 100 and up | | 100 and up | |

The exposed parts of the damper may be coated with a proper protecting agent to improve weather resistance.

The damper as the constituent elements of the anti-seismic device of the invention may have the following embodiments.

(1) One which is composed of a plurality of rigid hard plates and soft layers of viscoelastic material having the

material having the above-mentioned properties, with the external surface of the soft body covered with vulcanized rubber.

(4) One which is composed of a viscoelastic material having the above-mentioned properties and at least one skeleton of reticulated structure, corrugated structure, honeycomb structure, and woven stuff, with the exter-

nal surface of the soft body covered with vulcanized rubber. This embodiment may be produced as follows:

(4-1) By integrating under pressure the skeleton and viscoelastic material.

(4-2) By placing the skeleton and viscoelastic material alternately one over another.

(4-3) By placing the integral body of the skeleton and viscoelastic material (formed as in 4-1) and the skeleton and/or viscoelastic material alternately one over another.

(4-4) By combining the unit produced as in 4-1 to 4-3 with a plate or wire through lamination or any other proper means.

(5) The same embodiments as in 1-4 above, except that the external surface is not covered with vulcanized rubber.

The structure of the damper in the present invention is not limited to those of 1 to 5 above. For example, the covering rubber is not always necessary; instead, the damper may be held between two solid plates or enclosed in a container.

Another preferred embodiment of the damper used in the present invention is characterized by that at least one part thereof is made of slightly vulcanized rubber which is formed by vulcanizing a rubber compound incorporated with a vulcanizing agent in an amount equivalent to 1–70 wt % of the minimum amount used in common practice.

It is known well that a rubber material in the unvulcanized state exhibits an extremely high damping performance because it is an organic viscous body having a high molecular weight in such a state. Further, unvulcanized rubber exhibits a considerably high elastic modulus in the region of small deformation, as shown in FIG. 35; however, it rapidly decreases in elastic modulus when it undergoes large deformation (strain). For this reason, it lacks sufficient strength necessary for practical use. On the other hand, a rubber material comes to have a high elastic modulus and strength (which lead to a high restoring force) when it is sufficiently vulcanized (given a large number of crosslink points). Vulcanization, however, brings about a sharp decrease in damping performance (hysteresis loss).

The damper described herein as a preferred embodiment has both the high hysteresis (of unvulcanized rubber) and the outstanding mechanical properties (of vulcanized rubber), because it is made of rubber only slightly vulcanized with a small amount of vulcanizing agent.

The only slightly vulcanized rubber can be prepared from any rubber which can be vulcanized with a vulcanizing agent. Preferred examples of the rubber include ethylene-propylene rubber (EPR, EPDM), nitrile rubber (NBR), butyl rubber (IIR), halogenated butyl rubber (CIR), chloroprene rubber (CR), natural rubber (NR), isoprene rubber (IR), styrene-butadiene rubber (SBR), butadiene rubber (BR), acrylic rubber (AR), ethylene-vinyl acetate rubber (EVA), polyurethane (UR), silicone rubber (SiR), fluororubber (FR), chlorosulfonated polyethylene (CSM), and chlorinated polyethylene (CPE). They may be used alone or in combination with one another. To impart desirable hardness, loss characteristics, and durability according to the object of use, they may be incorporated with additives such as filler, tackifier, slip agent, anti xidant, plasticizer, softener, low-molecular weight polymer, and oil which are commonly used for rubber processing. Where the rubber is required to maintain its performance over a long period of time, it should be incorporated with a stabilizer (such as antioxidant, inhibitor, and antiscorch), or modified by hydrogenation or the like, so that it is stabilized. The above-mentioned additives can be used for stabilization.

The slight vulcanization is achieved by incorporating the rubber with a controlled amount of vulcanizing agent as explained in the following. In general, the optimum amount of vulcanizing agent (or the optimum crosslink density) is limited with some allowance for individual rubber compounds, so that the resulting vulcanized rubber has the necessary performance such as elastic modulus, strength, fatigue resistance, adhesion, and restoring force. On the other hand, the crosslink density of rubber is determined by the total amount of sulfur (or organic peroxide) and vulcanization accelerator (which are collectively referred to as "vulcanizing agent" hereinafter), because vulcanization is usually accomplished by using (1) sulfur as a major ingredient in combination with a vulcanization accelerator, (2) a small amount of sulfur and a large amount of vulcanization accelerator, or (3) an organic peroxide.

Rubber materials vary in physical properties depending on the vulcanizing (crosslinking) conditions. Commercial vulcanizing agents for a large variety of rubber compounds are shown in "Kogyo Zairyo" Vol. 29, No. 11 (1981), pp. 37–136 (published in Japan). According to this literature, individual rubber compounds are vulcanized with an average amount and minimum amount of vulcanizing agent as shown in Table 1. The amount is expressed in phr (parts by weight for 100 parts by weight of rubber).

TABLE 1

| Rubber | Average amount of vulcanizing agent (phr) | Minimum amount of vulcanizing agent (phr) |
|---|---|---|
| IR | 3.0 | 2.75 |
| SBR | 2.75 | 2.70 |
| BR | 2.4 | 1.30 |
| NR | 3.1 | 3.0 |
| NBR | 3.5 | 1.75 |
| CR | 1.0 | 0.8 |
| IIR | 3.0 | 2.75 |
| EPR | 3.5 | 2.72 |
| EPDM | 3.0 | 2.1 |
| AR | 1.5 | 1.0 |
| SiR | 0.75 | 0.55 |
| FR | 3.0 | 1.5 |
| CSM | 2.5 | 2.5 |
| CIR | 1.0 | 1.0 |
| CPE | 4.0 | 4.0 |
| UR | 8.0 | 3.0 |

CIR: halogenated butyl rubber

For a rubber compound to be made into a rubber product having a balanced performance, it should be incorporated with an optimum amount of vulcanizing agent, as mentioned above. This optimum amount corresponds to the average amount given in Table 1. On the other hand, "the minimum amount" in Table 1 applies to a special case in which the vulcanizing agent is used in a small amount purposely.

According to the present invention, the rubber shown in Table 1 should be incorporated with a vulcanizing agent in an amount equal to 1–70 wt %, preferably 5–60 wt %, and more preferably 10–50 wt % of the minimum amount.

Such a small amount of vulcanizing agent gives rise to the only slightly vulcanized rubber for the damper of the present invention, which has a hysteresis ($h_{50}$)

higher than 0.3, preferably higher than 0.35, and more preferably higher than 0.4 at 50% tensile deformation at 25° C.

The only slightly vulcanized rubber should have a glass transition temperature (Tg) outside the range of −10° C. to 30° C. and storage moduli (E) at −10° C. and 30° C. for repeated deformation of 0.01% at 5 Hz whose ratio (E at −10° C. to E at 30° C.) is lower than 10, preferably lower than 7, and more preferably lower than 5, and most desirably lower than 3.

The damper made of the only slightly vulcanized rubber has the structure as explained in the following with reference to the accompanying drawing.

The structure of the damper is classified into the following three groups.

(A) Simple structure made of the only slightly vulcanized rubber as the major constituent.

(B) Composite structure made of the only slightly vulcanized rubber and a hard material.

(C) Composite unit in which the body of structure (A) or (B) is combined with hard plates.

The simple structure (A) is further divided into the following four subclasses.

(A-1) Monolithic structure which is made of the only slightly vulcanized rubber alone. Examples of this structure are shown in FIG. 22 (rectangular prism) and FIG. 23 (cylinder). The shape of the product (damper) may be properly selected according to the object of use.

(A-2) Multilayered structure which is made of the only slightly vulcanized rubber of different kinds (in rubber type, compounding, and degree of crosslinking). The multiple layers may be arranged vertically, horizontally, or coaxially. The multilayered structure also includes the macroscopic uneven dispersion of rubber components of different kinds. With this structure, the damper can exhibit any desired performance (such as elastic modulus, failure characteristics, and hysteresis). Examples of this structure are shown in FIGS. 24 to 26. The dampers 103, 104, and 105 shown in these figures are made up of layers $a_1$, $a_2$, . . . $a_n$ arranged horizontally, vertically, or coaxially. The individual layers are made of the only slightly vulcanized rubber which varies in type of rubber, compounding, and degree of crosslinking.

(A-3) Composite structure which is made of the only slightly vulcanized rubber and highly vulcanized rubber. In this structure, the outside or inside of the only slightly vulcanized rubber is partly provided with "highly vulcanized rubber" (formed according to the average formulation shown in Table 1).

In the example shown in FIG. 27, the damper 106 is made up of the only slightly vulcanized rubber a (forming a core) and the highly vulcanized rubber b (forming a cover thereon). In the example shown in FIG. 28, the damper 107 is made up of the highly vulcanized rubber b (forming a lattice frame) and the only slightly vulcanized rubber a (filling the space of the lattice frame). In another example, the only slightly vulcanized rubber may container the highly vulcanized rubber dispersed therein.

The highly vulcanized rubber used for this structure may be incorporated with additives such as filler, tackifier, slip agent, antioxidant, plasticizer, softener, low-molecular weight polymer, and oil which are commonly used for rubber processing. Furthermore, the highly vulcanized rubber may be used in combination with hard materials (mentioned in (B) later) to form a laminate structure.



(A-4) Composite structure which is made of the only slightly vulcanized rubber and unvulcanized rubber. Unvulcanized rubber can be partly used as supplement although it cannot be used as the principal constituent in this invention as mentioned above. An embodiment of this structure may be made up of the only slightly vulcanized rubber and unvulcanized rubber dispersed therein. (In this case, the unvulcanized rubber may be replaced by the above-mentioned plasticizer, softener, tackifier, oligomer, or slip agent.) In an example shown in FIG. 29, the damper 108 is made up of the only slightly vulcanized rubber a and the unvulcanized rubber c enclosed therein.

The composite structure (B) includes those which are formed by combining the structure (A-1), (A-2), (A-3), or (A-4) with a hard material. In the dampers 109, 110, and 111 of composite structure shown in FIGS. 30 and 31, the layers of the only slightly vulcanized rubber a and the layers of hard material d are laminated horizontally or vertically on top of the other or arranged alternately coaxially. The hard material, which is not specifically limited, includes, for example, metal, ceramics, glass, FRP, plastics, polyurethane, hard rubber, wood, rock, sand, and leather. The hard material may be in the plate, reticulated, corrugated, honeycomb, or woven form.

The composite unit (C) is shown in FIG. 33. The damper 12 in this example is made up the body e of structure (A) or (B) and the hard plates f bonded to the top and bottom thereof. In actual application, one or more units of the damper 12 are arranged horizontally or vertically. Where two or more units are used, they may be of the same or different type and structure. The hard plates used in this structure include those of metal, ceramics, FRP, plastics, glass, wood, paper, polyurethane, and hard rubber.

In the case of the structure containing hard plates, the damper 113 shown in FIG. 34 is provided with a layer of highly vulcanized rubber b (of the same of different kind) to increase the bond strength between the only slightly vulcanized rubber and the hard plate. This layer may be replaced by a proper adhesive.

The anti-seismic device of the present invention is composed of dampers and anti-seismic rubber bearings arranged in parallel. The damper is constructed mainly of a viscoelastic material as mentioned above, and the anti-seismic rubber bearing is composed of rigid hard plates and viscoelastic soft boards laminated alternately one over the other.

The damper constituting the anti-seismic device of the present invention is not specifically limited in shape so long as it has a shape which produces the damping effect upon shear deformation or flexural deformation. In general, a cylindrical shape is desirable.

In the present invention, the dampers are used in combination with the anti-seismic rubber bearings. They may be arranged in parallel between a building and a foundation. Alternatively, the damper may be placed in a cylindrical space formed at the core of the anti-seismic rubber bearing.

The anti-seismic rubber bearing constituting the anti-seismic device of the present invention is composed of hard plates and soft boards. The hard plates may be made of metal, ceramics, plastics, FRP, polyurethane, wood, paper board, slate, and decorative laminate. The soft boards may be made of vulcanized rubber, unvulcanized rubber, plastics, rubber or plastics foam, asphalt, clay, and mixtures thereof. The hard plate and

soft board may have a shape of circle, square, pentagon, hexagon, or polygon.

The anti-seismic rubber bearing may be covered with a weather-resistant covering rubber for the improvement of weather resistance as mentioned above.

The anti-seismic device of the present invention constructed as mentioned above produces both the anti-seismic effect and damping effect, thereby absorbing and reducing the earthquake motion that is transmitted to the building at the time of earthquake.

The examples of the invention are now described with reference to the drawings.

First, an example of the damper used in the invention will be explained.

FIG. 1 is a longitudinal sectional view of the damper 1 pertaining to the above-mentioned embodiment 1. This damper 1 is composed of rigid hard plates 3 and soft layers of viscoelastic material 2 which are laminated alternately. The hard plates are steel plates or the like, and the viscoelastic material has the above-mentioned characteristic properties (i) and (ii). The damper is covered with the covering rubber 4 of vulcanized rubber. The hard plates 3*a* and 3*b* positioned at the top and bottom of the damper function also as the flanges.

The laminate structure just mentioned above produces a pronounced damping effect because the soft layers undergo the maximum deformation (shear deformation) when the damper receives vibration. Because of this damping effect, the damper of the invention is much smaller in size than the conventional oil-type viscous damper.

In designing the damper 1 of this example, it is possible to select any shape, volume ratio, and number of layers for the soft layers of viscoelastic material 2 and the hard plates 3, according to the spring constant and damping effect required under actual use conditions. The damper in the simplest structure is composed of two hard plates and one soft layer interposed between them.

The thickness of the covering layer 4 is properly selected according to the size and object of the damper. Usually it is greater than 1 mm and smaller than 100 mm.

The material for the hard plates 3 may be selected from metal, ceramics, plastics, FRP, polyurethane, wood, paper board, slate, and decorative laminate.

FIGS. 2 and 3 are longitudinal sectional views of the damper 1 pertaining to the above-mentioned embodiment 2. This damper 1 is composed of a soft body of viscoelastic material 2 and spherical bodies 5 or cylindrical bodies 6 of solid substance embedded in said soft body. The soft body is covered with the vulcanized rubber 4 and is held between the flanges 3*a* and 3*b*.

The solid material embedded in the damper permits the viscoelastic material 2 to greatly deform, providing the good damping effect.

FIG. 4 is a longitudinal sectional view of the damper 1 pertaining to the above-mentioned embodiment 3. This damper is composed of the soft body and the vulcanized rubber 4 covering it. The soft body is made of a viscoelastic material 2 alone and is held between the flanges 3*a* and 3*b*.

In the example shown in FIG. 2, the spherical body 5 is not necessarily required to be truly spherical, nor is it required to be uniform in size. Rather, a proper size distribution may be desirable.

The diameter (or average diameter) D of the spherical body 5 may vary depending on the size of the damper and the quality of the spherical body 5 and viscoelastic material 2. It is in the range of $0.1 \leq D \leq 10$ (mm), and preferably $1 \leq D \leq 10$ (mm).

The amount of the spherical bodies 5 should be such that the following relationship is established.

$$0.5 \leq \frac{V_R}{V_R - V_L} \leq 0.9 \text{ and preferably}$$

$$0.5 \leq \frac{V_R}{V_R - V_L} \leq 0.8$$

where $V_L$ is the volume of the viscoelastic material 2 and $V_R$ is the volume of the spherical bodies 5.

The solid substance may be the columnar body 6 having a round section as shown in FIG. 3. In addition, the solid substance may have a shape of spheroid or flat spheroid.

The spherical bodies 5 and columnar bodies 6 should be uniformly dispersed in the soft body. They may be hollow for the adjustment of specific gravity.

The solid substance may also be something like a wall which increases the area of contact with the viscoelastic material. In other words, it may be a partitioning member which separates the soft body into cells.

The preferred partitioning member is one which forms vertically elongated cells in the soft body of the damper as shown in FIGS. 6(*a*) to 6(*e*). FIGS. 6(*a*) to 6(*e*) are perspective views showing the partitioning members 8. The one shown in (a) is concentric, the one shown in (b) is radial, the one shown in (c) is a combination of the ones shown in (a) and (b), the one shown in (d) is cylindrical, and the one shown in (e) is spiral. The outermost cylinder in FIGS. 6(*a*) to 6(*e*) shows the internal wall (vulcanized rubber) of the damper.

The partitioning member may also be of honycomb structure. Such a structure should preferably be symmetrical with respect to the center axis of the soft body so that the stress is uniformly distributed at the time of earthquake.

The partitioning member shown in FIGS. 6(*a*) to 6(*e*) should preferably be fixed to the damper or flange; however, it is not always necessary that the edge that comes into contact with the flange and the edge that comes into contact with the internal wall of the vulcanized rubber of the damper be all fixed. For example, in the case of the one shown in FIG. 6, some of the partitioning members are fixed at the upper end alone and others are fixed at the lower end alone.

The solid substance from which the spherical bodies, columnar bodies, and cell partitioning members are made is not specifically limited. It includes, for example, metal, ceramics, glass, FRP, plastics, polyurethane, hard rubber, wood, rock, sand, and pebbles. In addition, the partitioning member may be made of rubber, paper, and leather having a comparatively low hardness.

Examples of the rubber material include vulcanized rubber exemplified above as the viscoelastic material. Preferred examples of the plastics include thermoplastic resins such as polystyrene, polyethylene, polypropylene, ABS, polyvinyl chloride, polymethyl methacrylate, polycarbonate, polyacetal, nylon, polyether chloride, polytetrafluoroethylene, acetyl cellulose, and ethyl cellulose; and thermosetting plastics such as phenolic resin, urea resin, unsaturated polyester resin, epoxy resin, alkyd resin, and melamine resin.

Preferred examples of the FRP may be filler- or fiber-reinforced rubber or plastics.

It is not always necessary that the solid substance is made of a single material; but it may be formed by the combination of the above-mentioned materials. Examples are a combination of metal and rubber and a combination of plastics and rubber.

The damper as shown in FIG. 1 may be produced by bonding with an adhesive the covering rubber 4, which has previously been vulcanized in the plate or film form, to the external surface of the laminate body composed of soft layers and hard plates. According to a preferred method, it is produced by bonding the covering rubber 4 in unvulcanized state to the external surface of the laminate body composed of soft layers and hard plate, and subsequently performing vulcanization, thereby bonding and integrating the covering rubber 4.

In this case, the vulcanization may be accomplished by heating in the usual way or by using electron rays, radiation, or ultrasonic wave. Low-temperature vulcanization can also be used.

In the case where sufficient adhesion or bonding is not made between the inner viscoelastic material and the covering rubber by the vulcanization bonding, an additional rubber layer which exhibits good adhesion to both of them may be interposed between them. In addition, for the improvement of adhesion, the internal viscoelastic material and/or covering rubber may be incorporated with an additive.

The vulcanized rubber 4 for covering is usually placed outside the laminate body composed of soft layers of viscoelastic material 2 and hard plates 3, as shown in FIG. 1. However, for the increased bonding between the covering rubber and the hard plate or for other manufacturing reasons, the edges of the hard plate 3 may project into the covering layer 4 as shown in FIG. 8.

The damper as shown in FIGS. 2 to 4 may also be produced in the same manner as mentioned above. In this case, it is desirable that a special pack be made in which the viscoelastic material and solid substance are enclosed and it be then inserted into the main body (covering rubber) of vulcanized rubber which has been separately prepared.

According to another method of production, a pack 13' in which are enclosed the viscoelastic material and solid substance is inserted into the main body 4 of vulcanized rubber through a hole 9 made in the flange 3*a* as shown in FIG. 7. After insertion, the hole is closed with a threaded stopper 9*a* which is screwed into the threaded hole 9. The pack may be fixed to the inside wall of the vulcanized rubber 4 and the flanges 3*a* and 3*b*, or may be simply inserted.

The material from which the pack 13' is made is not specifically limited. It includes, for example, rubber, polyurethane, plastics, FRP, paper, leather, and metal plate. Rubber, plastics, and FRP may be selected from those which are exemplified above for the solid substance.

It is not always necessary that the pack 13' be made of a single material; but it may be formed by the combination of the above-mentioned materials. Examples are a combination of metal and rubber and a combination of plastics and rubber.

In the case where the main body of the damper and the pack are produced independently from each other, the production is much easier and the production cost is much lower than in the case where they are produced as an integral body. In the former case, it is possible to replace the pack alone or the vulcanized rubber alone.

If the damper of the present invention is used under the conditions that cause considerable deformation, a great local strain would occur in that part of the soft layer and covering layer that is in contact with the edge of the hard plate (shown in FIG. 1). This local strain might lead to the breakage of the structure. To prevent this possible trouble, the hard plate 3 may be provided with a round edge (a) which projects into the covering vulcanized rubber 4, as shown in FIG. 9. In addition, the corner (b) formed by the covering vulcanized rubber 4 and the flange 3*b* may be filleted as shown in FIG. 9.

The damper of the present invention may be composed of at least one skeleton of reticulated structure, corrugated structure, honeycomb structure, or woven stuff and the viscoelastic material. In this case, the material from which the skeleton is made is not specifically limited. It includes, for example, metal, ceramics, plastics, FRP, polyurethane, natural fiber (cotton and silk), and synthetic fiber (polyamide and polyester).

The combination of the viscoelastic material with the skeleton as in the above-mentioned embodiment 4 gives an extremely high damping coefficient to the damper of the present invention.

The viscous viscoelastic material produces the stress-strain curve as shown in FIG. 10. Thus it absorbs stress when deformation is small; but it decreases in stiffness and hence does not have sufficient strength when deformation is large. By contrast, the reticulated structure, corrugated structure, honeycomb structure, and woven stuff produce the stress-strain curve as shown in FIG. 11. Thus they exhibit high stiffness when deformation is large.

The damper of combination structure, therefore, produces a good damping effect while keeping a high stiffness over a broad range from small deformation to large deformation.

In the case of the damper of this structure, the skeleton should be arranged such that the direction in which the skeleton deforms most easily is horizontal. For example, the skeleton of reticulated structure should preferably be arranged such that one diagonal line of each opening is horizontal.

The anti-seismic device of the invention will be described with reference to examples that follow.

FIG. 12 and FIG. 13 are respective longitudinal sectional views of the anti-seismic devices pertaining to the examples of the present invention.

The anti-seismic device shown in FIG. 12 is composed of the anti-seismic rubber bearings 10 and the dampers 1 arranged in parallel. The anti-seismic rubber bearing 10 is composed of a plurality of rigid hard plates 11 and soft layers 12 having the viscoelastic property laminated one over another. The damper 1 is as described above. In FIG. 12, there are shown flanges at 13–16, a building at 20, and a foundation at 30.

The anti-seismic device shown in FIG. 13 is of such a structure that the damper 1 is placed in the cylindrical space formed at the core of the anti-seismic rubber bearing 10. (In FIGS. 12 and 13, like reference numerals designate corresponding parts.)

Other embodiments relating to that shown in FIG. 13 are shown in FIGS. 36 and 37. The embodiment shown in FIG. 36 is a modification of the embodiment shown in FIG. 13. The modification is made by making a through hole at the center of the damper 1 shown in FIG. 13 and filling it with an elastoplastic body 91, a preferred example of which is lead. The embodiment

shown in FIG. 37 is a modification of the embodiment shown in FIG. 36. The modification is made by providing the flanges 13 and 14 with fasteners 92 and 93. The fastener 92 is open downward and the fastener 93 is open upward, so that they hold the upper and lower ends of the elastoplastic body 91.

In the case of the anti-seismic device as shown in FIG. 13, the surrounding vulcanized rubber for the anti-seismic rubber bearing functions as the covering rubber; therefore, it is not always necessary that the damper 1 be of such a structure that the soft body of viscoelastic material is covered with vulcanized rubber. In the case where the top and bottom of the viscoelastic material are covered with thick flanges 3*a* and 3*b*, respectively, as shown in FIGS. 1 to 4, it is not always necessary that the top and bottom of the damper be fixed to the building and foundation, respectively, with additional hard plates (such as steel plates) interposed between them.

In the case of the anti-seismic device shown in FIG. 12, the number of the anti-seismic rubber bearings 10 and dampers 1 and the intervals of their arrangement may be properly determined according to the object of using the anti-seismic device. In the case of the anti-seismic device shown in FIG. 13, the ratio of the sectional area of the anti-seismic rubber bearing 10 to the sectional area of the damper 1 may be properly determined according to the object of using the anti-seismic device.

The above-mentioned anti-seismic device of the present invention as shown in FIG. 12 is formed by arranging the anti-seismic rubber bearings 10 and the dampers 1 in parallel. The anti-seismic rubber bearing is produced by laminating hard plates and soft layers one over another and bonding them together with an adhesive or by covulcanization.

The anti-seismic device as shown in FIG. 13 is produced in the following manner. At first, the anti-seismic rubber bearing 10 is formed by vulcanization, with the core left void. Subsequently, the previously molded damper 1 is inserted into the core void, or hard plates and soft layers are placed in the core void alternately and the resulting assembly is subjected to covulcanization.

The anti-seismic device having a built-in damper as shown in FIG. 13 exhibits good damping characteristics over a broad range of small deformation to large deformation; however, there is an instance where it produces only a little damping effect for very small vibration.

The anti-seismic rubber bearing, with its core filled with a damper of viscoelastic material having good hysteresis characteristics, provides an extremely high damping factor over a broad range of small deformation to large deformation. If the specific viscoelastic material is to exhibit a high damping capacity, it will have a higher elastic modulus than the surrounding anti-seismic rubber bearing in the range of extremely small deformation. Needless to say, the ratio of increase is by far smaller than that of lead or steel plastic bodies. In any way, high moduli are generally inevitable where the high loss characteristics are required.

In the present industrial society, the countermeasure for microvibrations is required by IC plants, biotechnology plants, and laser plants where accurate processing is necessary and also by houses along a railway line or a motorway. The ordinary anti-seismic rubber bearings are effective against microvibrations and produce good damping effect because they have a low modulus in the lateral direction. By contrast, the anti-seismic rubber bearing containing a viscoelastic material has a higher modulus than the ordinary anti-seismic rubber bearing without the viscoelastic material, because the viscoelastic material has a high modulus for small deformation. Therefore, the anti-seismic rubber bearing containing a viscoelastic material tends to be poor in damping effect for microvibrations and hence does not meet the present requirement.

In order that the damper does not prevent the damping action of the anti-seismic rubber for microvibrations, the anti-seismic device of the present invention should preferably have a structure as shown in FIG. 14.

The anti-seismic device as shown in FIG. 14 is composed of the anti-seismic rubber bearing 10 and the damper 1, with the space between them filled with a material 18 less resilient than the damper. The anti-seismic rubber bearing 10 is composed of a plurality of rigid hard plates 11 and soft layers 12 having the viscoelastic property placed one over another. The damper 1 is made mainly of a viscoelastic material. The damper 1 is placed in the space formed at the core of the anti-seismic rubber bearing 10. In FIG. 14, there are shown flanges at 13 and 14, a building at 20, and a foundation at 30.

The low-resilience material 18 should be one which satisfies the following condition.

$$\frac{E_L}{E_V} \leqq 0.9$$

$$\text{preferably } \frac{E_L}{E_V} \leqq 0.7$$

$$\text{more preferably } \frac{E_L}{E_V} \leqq 0.5$$

where $E_L$ is a storage modulus of the low-resilience material 18 and $E_V$ is a storage modulus of the viscoelastic material of the damper 1, both measured dynamically at 25° C., at a frequency of 5 Hz, and at a strain of 0.01%.

The low-resilience material 18 is not specifically limited so long at it satisfies the above-mentioned requirements. It may be made of a variety of rubbers and plastics and some of the above-mentioned viscoelastic materials. It may also be a spring of foam, metal, plastics, and rubber; blanket, woven stuff, and straw waste. The layer of the low-resilience material may has a space therein, according to need.

In addition, it is not always necessary that the low-resilience material 18 cover the side, top, and bottom of the damper 3. An embodiment may be possible in which the side alone is covered. It is desirable that the low-resilience material 18 be placed in the void between the damper 1 and the rubber laminate 10, as shown in FIG. 14. However, it is not always necessary to fill the void completely, and there may be some parts which are not filled with the low-resilience material.

In the anti-seismic device as shown in FIG. 14, there is no limitation in the shape of the anti-seismic rubber bearing, the size of the damper 1, and the thickness of the low-resilience material 18. They may be properly selected according to the object of using the anti-seismic device. For example, the anti-seismic rubber bearing 10 should preferably have a size which is defined as follows:

$$\frac{l}{L} \leqq 0.80$$

$$\text{preferably } \frac{l}{L} \leqq 0.70$$

$$\text{more preferably } \frac{l}{L} \leqq 0.64$$

where l is the diameter of the void in the anti-seismic rubber bearing **10**, and L is the diameter of the anti-seismic rubber bearing **10**. Also, the low-resilience material **18** should preferably have a size which is defined as follows:

$$\frac{l_0}{l} \leqq 0.1$$

$$\text{preferably } \frac{l_0}{l} \leqq 0.05$$

where $l_0$ is the thickness of the low-resilience material **18**, and l is the diameter of the void of the anti-seismic rubber bearing **10**.

The anti-seismic device just mentioned above may be produced by laminating hard plates and soft layers one over another and vulcanizing the assembly, with the core left void, and then inserting the previously made damper and low-resilience material into the void. According to another method, hard plates and soft layers, each having a hole at the center, are placed one over another around the previously formed damper and low-resilience material, and the resulting assembly is subjected to covulcanization.

The anti-seismic devices as shown in FIGS. **15** to **18** are composed of the anti-seismic rubber bearing **10** and the damper **1** placed in the center void thereof, with a space or an air-containing layer interposed between them. The anti-seismic rubber bearing **10** is composed of a plurality of rigid plates **11** and a plurality of soft layers **12** having the viscoelastic property placed one over another. The damper is made mainly of a viscoelastic material. The space or air-containing layer reduces to an extreme extent the amount of elastic deformation to be transmitted to the anti-seismic rubber bearing **10** when the damper **1** undergoes elastic deformation. Thus it enhances the damping action for microvibrations of the device.

In the anti-seismic device shown in FIG. **15**, the space **40** between the damper **1** and the bearing **10** is formed by a long member **41** (such as wire and rope) wound round the damper **1**. In the anti-seismic device shown in FIG. **16**, the space **40** is formed by the projecting parts **42** of the soft plates **12**, said projecting parts jutting inward the center void from the inside wall of the bearing **10**. In the anti-seismic device shown in FIG. **17**, the space **40** is formed by spherical bodies **43** arranged around the damper **1**. In the anti-seismic device shown in FIG. **18**, the space **40** is formed by sliding plates **44** attached to the top and bottom of the damper **1**.

The above-mentioned long member **41** and spherical body **43** may be replaced by a net, closed-cell foam, sheet having openings **45** as shown in FIG. **19**, and sheets **47** and **48** having ridges (or hollow ridges **49**) and grooves as shown in FIGS. **20** and **21**. The material to be arranged around the damper **1** should have a smaller compression modulus of elasticity per unit area in contact with the damper than that of the damper. The material should preferably be unvulcanized rubber, vulcanized rubber, polymer, fiber reinforced plastics, asphalt, clay, natural fiber, and metal. Preferable among them are vulcanized rubber and plastics.

As mentioned above, the damper used in the anti-seismic device of the present invention is constructed mainly of a viscoelastic material having a specific property. Consequently, it has the following advantages over the conventional viscous damper which employs oil.

(1) It is possible to select the temperature dependence and rate dependence of the hysteresis loss characteristics according to the properties of the individual rubber materials.

(2) Easy molding.

(3) Easy handling and easy execution.

(4) Easy maintenance.

(5) Low cost.

(6) High damping effect and small size.

The damper used for the anti-seismic device of the present invention does not have the disadvantage of the plastic damper but has much better characteristic properties than the conventional viscous damper. Therefore, it is of great industrial use.

The anti-seismic device of the present invention is composed of the dampers and the anti-seismic rubber bearings which are formed by laminating alternately a plurality of rigid hard plates and soft layers having viscoelastic properties. Therefore, the anti-seismic device produces both the anti-seismic effect and the damping effect and absorbs much of shaking at the time of earthquake, isolating the building from earthquake motion. Thus it prevents the building from crashing against other structures and also prevents the utilities such as water pipe, gas pipe, and wiring from damage at the time of earthquake.

In addition, it is expected that the anti-seismic device of the present invention produces the pronounced effect of removing, preventing, and suppressing vibrations.

The present invention differs from that disclosed in U.S. Pat. No. 4,566,231 relating to a vibration damping stiffener which produces a marked damping effect in the high-frequency region but produces almost no damping effect in the low-frequency region necessary for the anti-seismic structure. This is apparent from the experiment mentioned below.

This USP gives in FIGS. **7** and **8** the data of dynamic vibration response (acceleration vs. frequencies) observed with or without the stiffener. The difference ($\Delta G$) in the two accelerations is attributable to the damping effect of the stiffener. It was calculated from the formula $(\Delta G)=G_8-G_7$ (where $G_7$ is the acceleration in FIG. **7** which is obtained when the damper is not used, and $G_8$ is the acceleration in FIG. **8** which is obtained when the stiffener is used). $G_7$ and $G_8$ are shown in FIG. **38** which was drawn according to the data given in FIGS. **7** and **8** in said U.S. Pat. No. 4,566,231. The data of $\Delta G$ obtained from $G_7$ and $G_8$ shown in FIG. **38** are given in FIG. **39**. (FIG. **38** has the logarithmic ordinate and FIG. **39** has the linear ordinate.) It is noted from FIG. **39** that the stiffener disclosed in the USP produces the maximum damping effect at 244 Hz and the second largest damping effect in the neighborhood of 1000 Hz, especially at 1200 Hz.

The data apparently indicate that the stiffener disclosed in the USP produces almost no damping effect at frequencies under 100 Hz and produces completely no damping effect at earthquake frequencies of 1 to 10 Hz,

especially 4 to 5 Hz. This suggests that the stiffener will not effectively function when used as an anti-seismic device. The fact that the earthquake frequencies are mostly in the range of 1 to 10 Hz, especially 4 to 5 Hz, is clearly indicated in the following literature.

SEISMIC RESPONSE OF LIGHT INTERNAL EQUIPMENT IN BASE ISOLATED STRUCTURES by James M. Kelly and Hsiang-Chuan Tsai (Report No. UCB/SESM-84/17), Department of Civil Engineering, University of California, Berkeley, Calif., September 1984. (Figures in page 58).

In the meantime, the rubber compositions disclosed in U.S. Pat. No. 4,050,665 and U.S. Pat. No. 4,483,426 were tested for their characteristic properties. The results are shown below. It is noted that the hysteresis ratio ($h_{50}$) is much lower than that in the present invention.

| | U.S. Pat. No. 4,050,655 | U.S. Pat. No. 4,483,426 |
|---|---|---|
| Composition (parts by weight) | Polydimethylsiloxane 80<br>Silica 20<br>Dicyclopentadiene 0.5 | Butyl rubber 100<br>GPF carbon 33<br>Zinc oxide 5<br>Vulcanizing agent and others 8 |
| Breaking strength (kg/cm$^2$) | 26 | 44 |
| Elongation at break (%) | 500 | 470 |
| Hysteresis ratio ($h_{50}$) | 0.13 | 0.19 |
| Hd | 32 | 49 |

The specification refers to the disclosure of application Ser. No. 078,621 filed on July 28, 1987.

What is claimed is:

1. An anti-seismic device which comprises anti-seismic rubber bearings and dampers arranged in parallel, said anti-seismic rubber bearing being formed by laminating a plurality of rigid hard plates and soft boards having a viscoelastic property one over another, said damper being composed mainly of a viscoelastic material having the physical properties (i) and (ii) defined below.
   (i) the hysteresis ratio ($h_{50}$) is greater than 0.3 at 50% tensile deformation at 25° C.
   (ii) the storage modulus (E) measured dynamically at a frequency of 5 Hz, a strain of 0.01%, and a temperature of 25° C. is in the range of $1 \leqq E \leqq 2 \times 10^4$ (kg/cm$^2$).

2. An anti-seismic device as claimed in claim 1, wherein the viscoelastic material has an elongation higher than 1% at tensile break.

3. An anti-seismic device as claimed in claim 1, wherein the viscoelastic material is unvulcanized rubber or vulcanized rubber, or resin or plastic material having the above-defined characteristic properties.

4. An anti-seismic device as claimed in claim 3, wherein the unvulcanized rubber is one which has a Mooney viscosity $ML_{1+4}$ greater than 10 at 100° C.

5. An anti-seismic device as claimed in claim 1, wherein the damper is a laminate flexible body composed of a plurality of rigid hard plates and soft layers of viscoelastic material interposed between the rigid hard plates.

6. An anti-seismic device as claimed in claim 1, wherein the damper is a flexible body composed of a viscoelastic material and a solid substance embedded in said viscoelastic material.

7. An anti-seismic device as claimed in claim 1, wherein the damper is a flexible body composed of a viscoelastic material alone.

8. An anti-seismic device as claimed in claim 1, wherein the damper is a flexible body composed of at least one skeleton of reticulated structure, corrugated structure, honeycomb structure, and woven stuff, and a viscoelastic material.

9. An anti-seismic device as claimed in claim 5, wherein the flexible body is covered with vulcanized rubber.

10. An anti-seismic device as claimed in claim 1, wherein the flexible body is covered with a rubber material having good weather resistance.

11. An anti-seismic device as claimed in claim 1, wherein the anti-seismic rubber bearing has a void in which the damper is placed.

12. An anti-seismic device as claimed in claim 11, wherein the damper is arranged in the void of the anti-seismic rubber bearing with a material less resilient than the damper interposed between them.

13. An anti-seismic device as claimed in claim 1, wherein the damper is arranged in the void of the anti-seismic rubber bearing, and space or a layer which contains air is arranged between the damper and the anti-seismic rubber bearing.

14. An anti-seismic device as claimed in claim 11, wherein the damper has a hole extending in the same direction as said void and the hole is filled with an elastoplastic body.

15. An anti-seismic device as claimed in claim 14, wherein the elastoplastic body is lead.

16. An anti-seismic device as claimed in claim 14, which has flanges which hold between them the anti-seismic rubber bearing, damper, and elastoplastic body at both ends of the axis in which said void extends.

17. An anti-seismic device as claimed in claim 16, wherein said elastoplastic body is fixed by the holders attached to the flanges.

18. An anti-seismic device as claimed in claim 1, wherein said hysteresis ratio ($h_{50}$) is 0.35 or above.

19. An anti-seismic device as claimed in claim 1, wherein said hysteresis ratio ($h_{50}$) is 0.4 or above.

20. An anti-seismic device as claimed in claim 1, wherein at least part of said damper is made of only slightly vulcanized rubber which is formed by vulcanizing a rubber compound incorporated with a vulcanizing agent in an amount equal to 1 to 70 wt % of the minimum amount in common practice.

* * * * *

# United States Patent [19]

**Anderson**

[11] **Patent Number:** **4,566,678**

[45] **Date of Patent:** **Jan. 28, 1986**


[54] **POLYMERIC APPARATUS AND METHOD OF MAKING THE SAME**


[75] Inventor: **David G. Anderson,** Chesterton, Ind.

[73] Assignee: **Miner Enterprises,** Geneva, Ill.



[21] Appl. No.: **412,119**

[22] Filed: **Aug. 27, 1982**

[51] **Int. Cl.⁴** ........................ **B60F 11/00; B29C 17/00**

[52] **U.S. Cl.** ................................ **267/141.1;** 264/249; 264/301; 301/63 PW

[58] **Field of Search** ............... 264/294, 295, 296, 235, 264/346, 320, 322, 325, 326, 249; 267/153, 140, 63 R, 141, 141.1; 213/7, 22; 301/5.3, 63 PW



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*Primary Examiner*—Willard E. Hoag
*Attorney, Agent, or Firm*—Wood, Dalton, Phillips, Mason & Rowe



[57] **ABSTRACT**

A method of producing useful hollow bodies and the resulting products. The hollow bodies are formed by providing a block of thermoplastic elastomer material having a selected initial axial height and providing the block with a selected axial core opening. An axial force is applied to said block sufficient to compress said block a substantial extent to reduce the free height of the block and expand the core opening transversely outwardly to define sidewalls for the hollow body. The axial force is removed, and the hollow body is prepared for use as a compression spring or other useful device.


**34 Claims, 14 Drawing Figures**

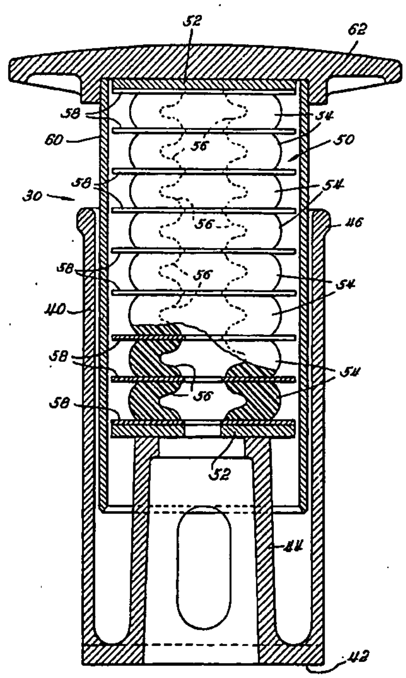


Fig.1

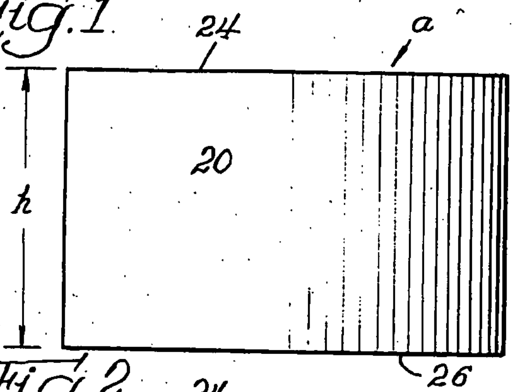



Fig.2

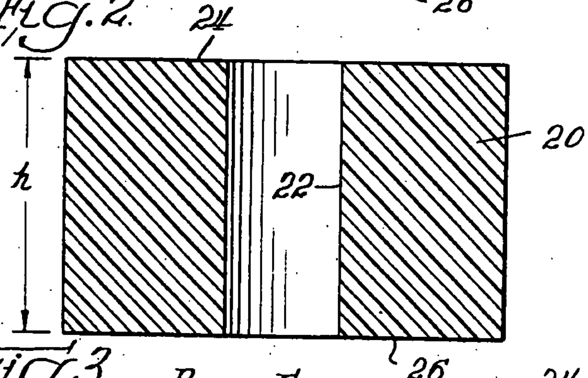



Fig.3

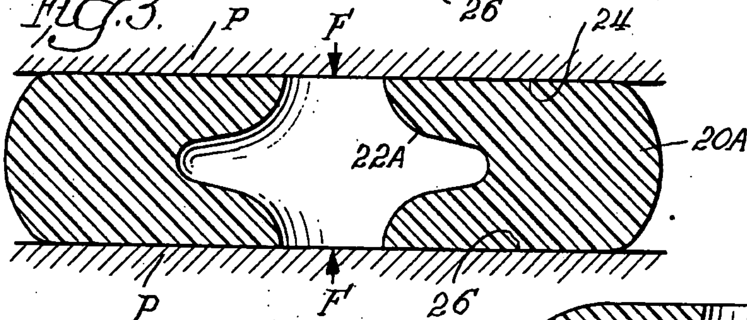
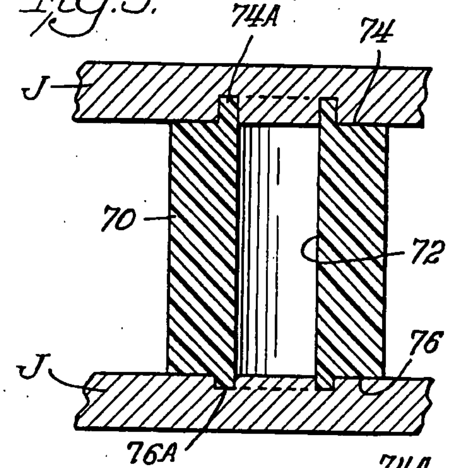



Fig.5



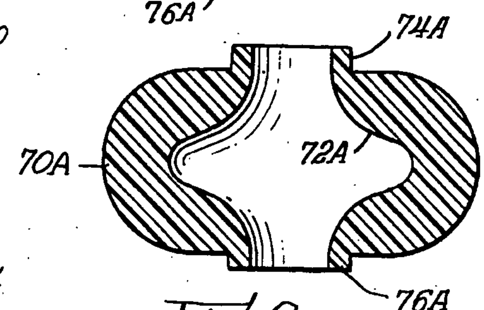




Fig.6

Fig.10

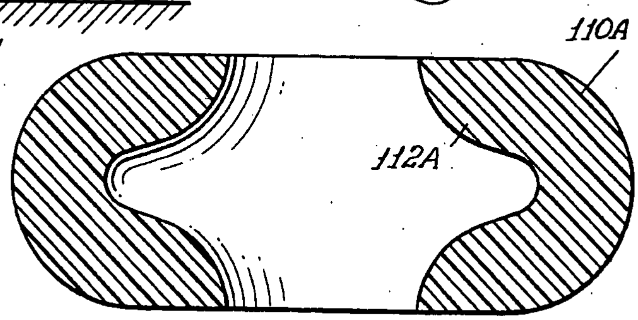



Fig.9

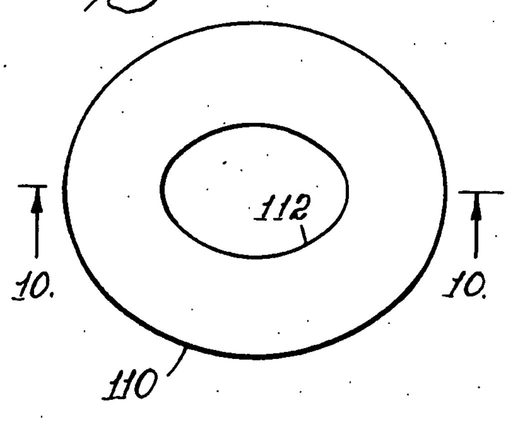



Fig.11

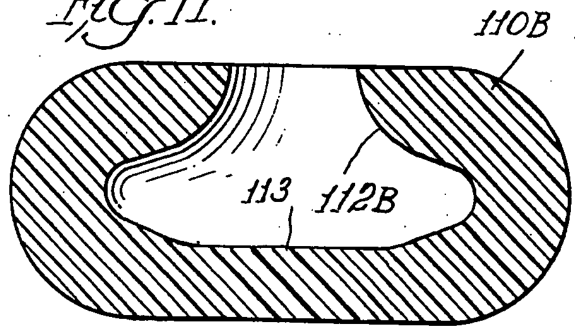


Fig. 4

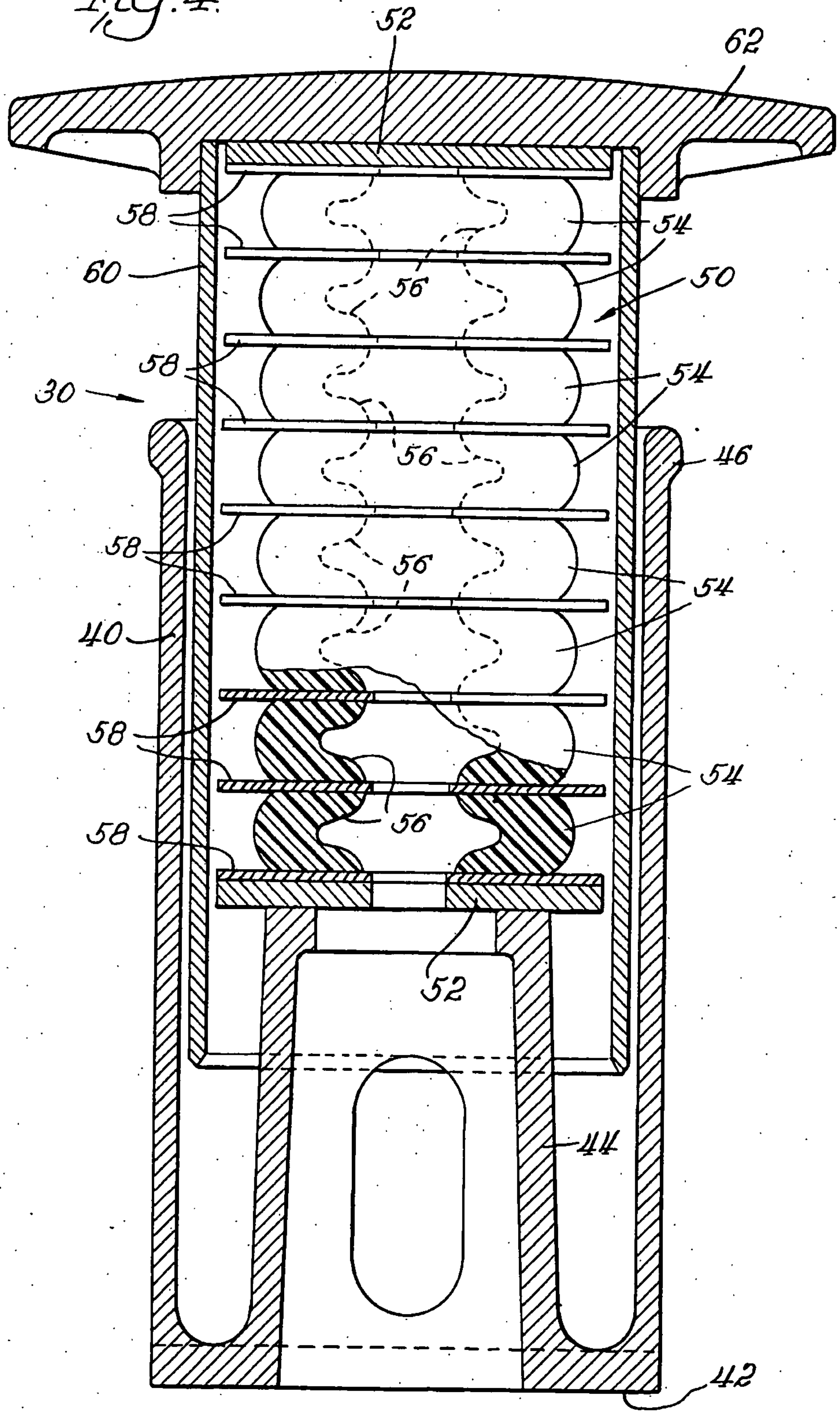
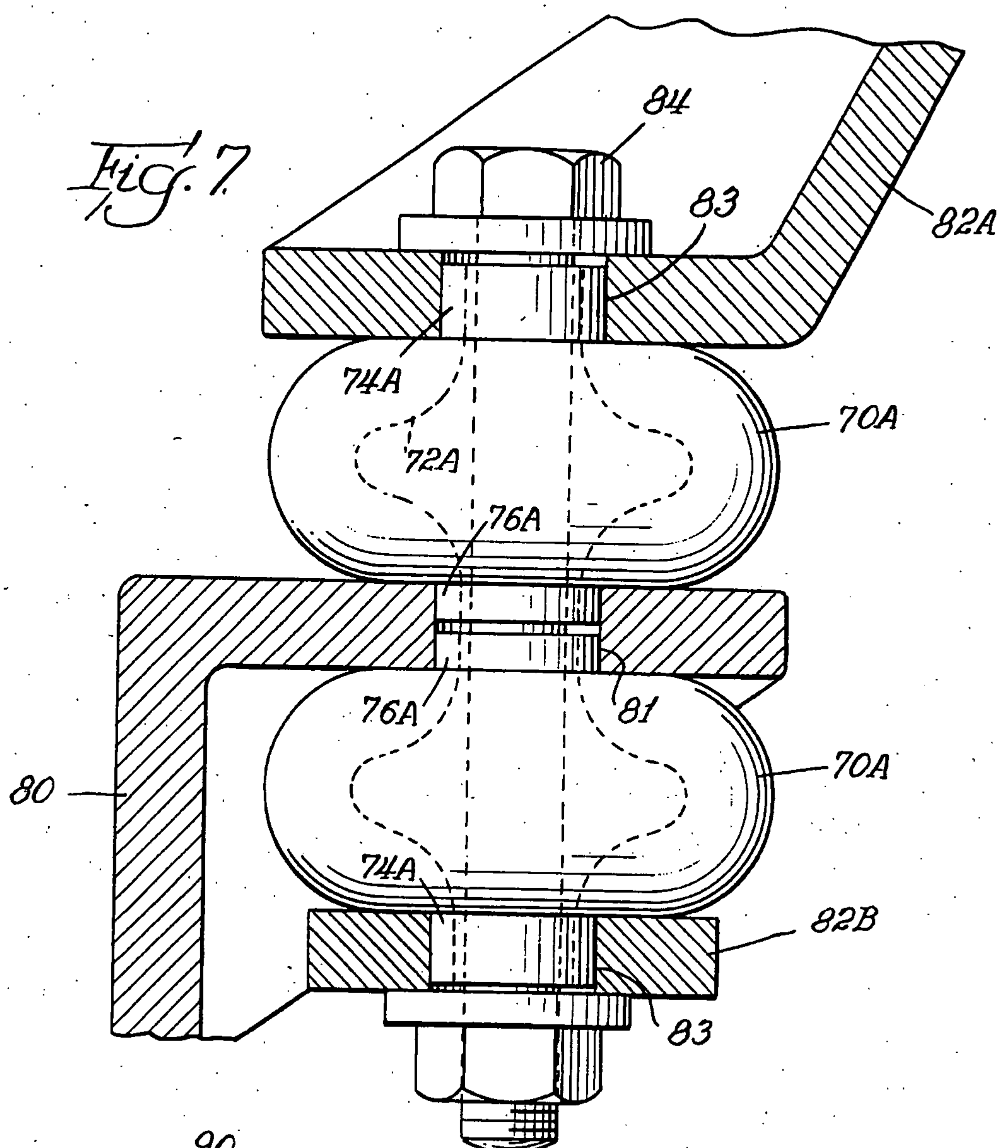


Fig. 7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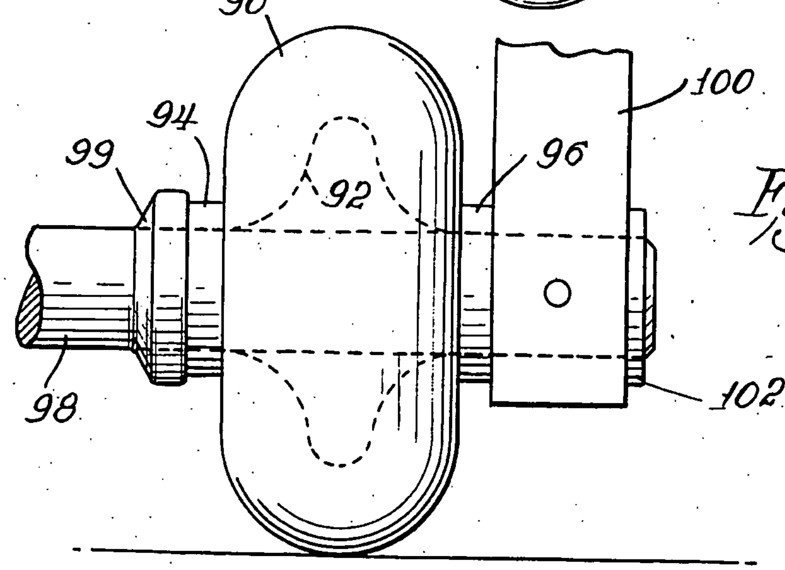
76A
81
70A
80
74A
82B
83

Fig. 8


90
100
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96
92
98
102

Fig. 12.

RUBBER SPRINGS

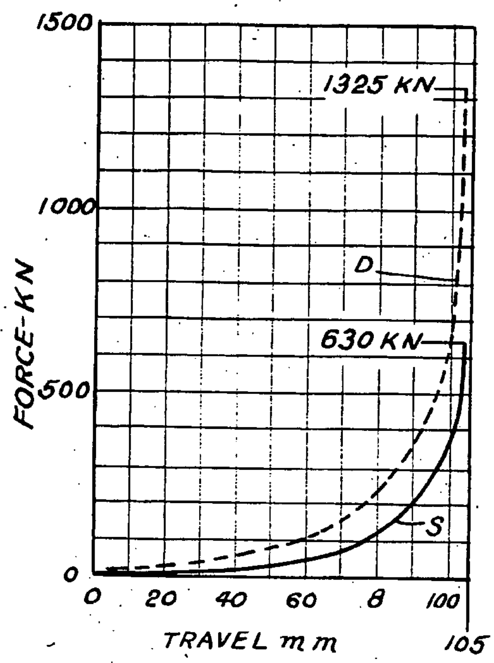
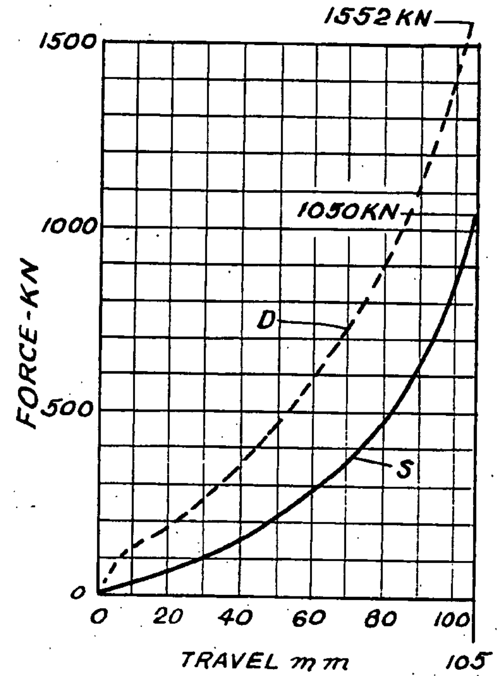


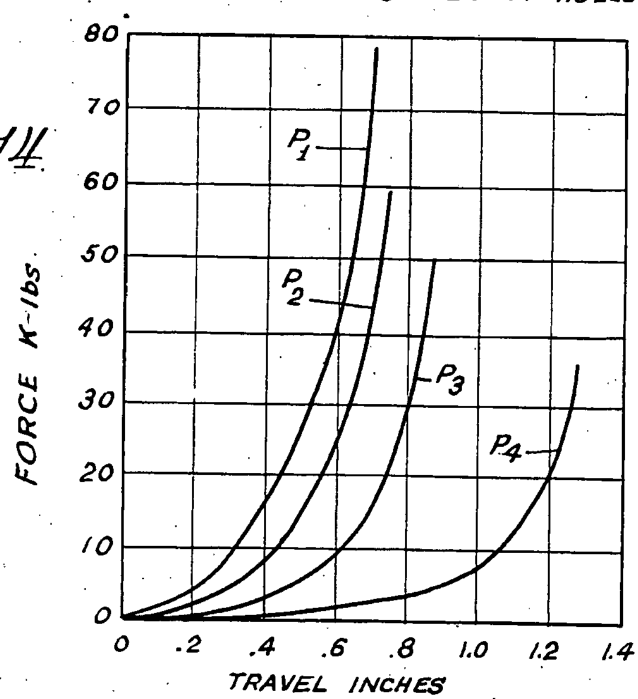

Fig. 13.

POLYMER SPRINGS




POLYMER SPRINGS WITH DIFFERENT HOLLOW CORES

Fig. 14.



# POLYMERIC APPARATUS AND METHOD OF MAKING THE SAME

## BACKGROUND AND GENERAL DESCRIPTION

This invention relates generally to an improved method of making useful hollow bodies of thermoplastic material, and more particularly relates to making hollow bodies from copolyester polymer elastomer material, and the products resulting therefrom.

As discussed in U.S. Pat. No. 4,198,037, issued on Apr. 15, 1980 to the assignee of the present application, elastomers have been widely used in the past for various products, including springs. One of the more recent useful thermoplastic elastomers is a copolyester polymer elastomer such as sold by E. I. duPont de Nemours & Co. of Wilmington, Del., under the trademark HYTREL. As explained in said patent, HYTREL is made from three ingredients, namely, diemethyl terephthalate; polyglycols such as polytetramethylene ether glycol, polyethylene ether glycol or polypropylene ether glycol; and short chain diols like butanediol and ethylene glycol. Generally, this product can be used to form thermoplastic elastomeric products. Similar elastomers may be produced and sold by others.

Generally, this type of polymer elastomer material has inherent physical properties that make it unsuitable for use as a compression spring. However, the recently issued patent referred to above describes a method by which the copolyester polymer elastomer material can be treated for rendering the material usable as a compression spring. Generally, that treatment, to convert the elastomer into compression spring material, comprises the application of a compressive force to a body of material which compresses the body in an axial direction to an extent greater than 30% of its previous axial length, measured in the direction of the applied pressure.

In continuing with the experimentation and development of the compression spring material disclosed in said patent, it has been found in accordance with the present invention that a physical modification to the elastomer block before the application of a compressive force thereto has an appreciable and unexpected effect on the physical characteristics, function and utilization of the final elastomer product. Generally, it has been found pursuant to this invention that the provision of a central core or opening in the copolyester polymer elastomer body, before the application of an axial compressive force to the body, has substantial beneficial effects on the resultant product. The use of the resulting hollow elastomer body as a compression spring is enhanced by changing the spring characteristics of the body and enlarging the scope of applications where such a spring can be utilized. The invention allows the physical characteristics of the hollow body to be varied easily to accommodate the loads and deflections needed in particular spring applications. Elastomer springs with various spring rates thus can be easily produced by using this invention. The resulting hollow elastomer body also possesses characteristics which make it useful in applications other than as a compression spring.

The provision of a core opening extending at least partially through the elastomer body before the application of a compression force has been found not to cause the sidewalls of the body to collapse, as may be expected. Rather, the sidewalls of the body and the core opening expand outwardly in a transverse direction as a result of the application of the compressive force to the body. The resulting sidewalls are generally uniform in thickness and symmetrical about the axial center-line of the core opening and the core opening has been greatly enlarged to define an essentially hollow body from the elastomer material. The presence of the core thereby changes the physical characteristics of the elastomer body.

The functional characteristics of the elastomer body are also changed by compressing the material with a core opening extending at least partially therethrough. When the body is utilized as a compression spring, the spring characteristics of the hollow body have been changed, compared to a solid body of the same material. The spring rate is changed, and the amount of dynamic and static energy which can be stored by the spring has been varied. The functional characteristics of the hollow elastomer bodies produced pursuant to this invention thus expands the flexibility of design and the scope of application for spring units utilizing copolyester polymer elastomer materials.

Moreover, the operating characteristics of the hollow elastomer body produced by this invention can be varied in a simple manner by changing the shape and size of the core opening provided in the body before compression. For example, if a lighter spring with a higher spring deflection rate per unit load is desired, the size of the core opening can be enlarged to correspondingly reduce the thickness of the resulting sidewalls of the hollow body. Similarly, a stiffer spring can be produced by selecting a smaller core opening so that the increased sidewall thickness produces a stiffer spring. The shape of the elastomer body, as well as the shape of the central core opening can also be varied to suit particular applications. For example, the body can be cylindrical, oval, generally rectangular or square in configuration. The core opening likewise can be circular, oval, rectangular or square in cross-section. In the preferred embodiment the transverse shape of the core opening and the body are similar and co-axial, so that the symmetry of the body is maintained.

In addition, the hollow bodies in accordance with this invention are simple and economical to produce, compared to hollow bodies made by prior techniques. Instead of requiring expensive and cumbersome cores or internal machining to produce a hollow opening in a body, the present invention requires only the provision of a core opening in the elastomer body prior to compression. This opening easily can be drilled into or through the body with conventional drilling equipment. Alternately, if the elastomer body is to be molded, the central opening can be molded into the body prior to compression. If the initial elastomer body is to be cylindrical, for example, a simple pipe section can be used for the mold for the body as well as for the core opening.

Also, relatively minor modifications in the jigs and fixtures used to produce the hollow bodies will result in variations in the shape of the final product. For example, the jigs and fixtures used to shape the hollow body can be arranged to provide one or both axial ends of the body with a reduced collar or neck portion which could be used to mount and position the body in particular industrial applications. Similarly, the jigs and fixtures can be arranged so that the final core opening extends only partially through the body. One axial end of the body will thereby be provided with a solid end wall that is desirable for particular applications.

The hollow elastomer bodies produced in accordance with this invention also can be designed and used for purposes other than industrial compression springs. They are useful for example, as isolation and vibration dampers, such as in motor mount applications. They are also useful as energy absorption bumpers or cushions. The hollow elastomer bodies produced pursuant to the invention also have sufficient symmetry about their axial dimensions to be utilized in various applications as rollers or wheels.

## DESCRIPTION OF ILLUSTRATIVE EMBODIMENTS

Further advantages and features of the present invention will become more apparent from a description of several embodiments thereof taken in conjunction with the accompanying drawings in which:

FIG. 1 is a front elevational view of a cylindrical block of copolymer polyester elastomer material which can be used to produce a hollow elastomer body in accordance with this invention;

FIG. 2 is a front cross-sectional elevational view of the elastomer body shown in FIG. 1, illustrating a central axial core opening provided in the body, and also illustrating the condition of the body before the application of an axial compressive force;

FIG. 3 is a cross-sectional front elevational view of the elastomer body shown in FIGS. 1 and 2 depicted in the process of having an axial compressive force applied to the body pursuant to this invention;

FIG. 4 is a cross-sectional front elevational view of a freight and locomotive buffer including a compression spring assembly formed from a stack of hollow elastomer bodies produced pursuant to this invention;

FIG. 5 illustrates a modification of the invention wherein the hollow body is provided with a reduced neck portion at both axial ends, and a central core opening, shown before a compression force is applied and further showing the body positioned between compression members;

FIG. 6 is a cross-sectional elevational view of the modified hollow body shown in FIG. 5 after the application of a compressive force to the body;

FIG. 7 is a partial cross-sectional elevational view of a motor mount assembly utilizing a pair of aligned hollow spring bodies conforming to the modified embodiment shown in FIGS. 5 and 6;

FIG. 8 is a partial view of a further embodiment of the present invention showing the hollow elastomer body used as a wheel;

FIG. 9 illustrates a further modification of a hollow elastomer body in accordance with this invention where the body and the central opening are oval in horizontal cross-section;

FIG. 10 is a cross-sectional elevational view of the oval elastomer body shown in FIG. 9, taken along the major axis depicted by the line 10—10 in FIG. 9 after the body is subjected to the compressive force in accordance with this invention;

FIG. 11 is a cross-sectional elevational view of a modified oval body after compression, showing the final configuration of the oval body when the core opening extends only partially through the body so that a solid wall is formed at one axial end;

FIG. 12 is a force-travel diagram illustrating the results of static and dynamic tests on a rubber elastomer body having the general configuration of the body illustrated in FIG. 1;

FIG. 13 is a force-travel diagram illustrating the results of static and dynamic tests on a hollow copolyester polymer elastomer body pursuant to this invention having the general configuration of the body illustrated in FIG. 1; and

FIG. 14 is a force-travel diagram comparing the results of applying a selected compressive force to cylindrical bodies of copolyester polymer elastomer having the same initial size and shape but having a central core of varying sizes.

The method of producing improved elastomer bodies pursuant to this invention will be described initially by reference to the cylindrical form of hollow body illustrated in FIGS. 1-3. As shown therein, a cylindrical body 20 of copolyester polymer elastomer material has a selected initial axial height 'h' and a circular transverse cross-sectional area 'a'. As shown in FIG. 2, the body 20 is also provided with a central core opening 22. The core 22 may be a drilled circular opening which extends axially through the body 20 from the top end wall 24 of the body to the bottom end wall 26. Pursuant to this invention, the body 20 having the core opening 22 is placed within a suitable compression jig, such as illustrated by the plates P in FIG. 3.

As explained in said patent, the block of polymer in the preferred embodiment is then annealed. The annealing time could extend to about one-hundred and eighty hours for particular applications. Next, an axial force 'F' is applied to the body 20 to compress the body a selected axial extent. The force 'F' should be sufficient to compress the elastomer body 20 at least 30% of its initial axial height 'h'. The optimum results occur when the force 'F' compresses the body 20 by approximately 50% of its initial height 'h'. An operative range of forces would cause compression in the range of 30% to about 80% of the original axial height 'h' of the body 20. As explained further in U.S. Pat. No. 4,198,037, this compressive force 'F' changes the spring-related properties of the elastomer and permits the body 20 thereafter to be used as a compression spring with elastic memory.

The result of the above-described application of force 'F' to produce a hollow body 20A having the advantages and characteristics of the present invention is illustrated in FIG. 3. The compression of the body 20 not only changes the physical characteristics of the polymer material, but it transversely and outwardly expands the core opening 22, to produce a generally toroidally shaped hollow body 20A having an enlarged symmetrical core 22A, as illustrated in FIG. 3. This hollow body 20A has different physical characteristics as compared to a solid body of the same elastic material subjected to the same compressive force. As also illustrated in FIG. 3, the resultant hollow body 20A has uniform sidewalls, and is symmetrical about its axial center line. The body 20A is thereby useful as a compression spring, a vibration dampener, an energy absorption cushion, or as a hollow rotary member, such as a wheel or the like.

FIG. 4 illustrates that the resulting hollow elastomer bodies, such as illustrated in FIG. 3, can be utilized to form a compression spring assembly in a freight and locomotive buffer 30. These buffers 30 are typically used between railway cars to buffer the impact of adjacent cars, and to compensate for the impact loads on the car couplers during operation of the freight train. To accomplish these purposes, the buffer 30 includes a housing 40 which has a flat rear mounting wall 42. The wall 42 is adapted to be mounted on a freight car in the

desired location where the impact or shock energy must be absorbed by the buffer 30. Extending inwardly from the wall 42 is a central block 44 which supports a spring assembly 50. As described hereinbelow, the housing 40 is typically cylindrical, and preferably has an enlarged outer rim 46.

The buffer 30 also includes a sliding inner cylinder 60. This cylinder 60 is telescopically arranged within the housing 40, as shown in FIG. 4, and includes a striker head 62 at its outer end. The cylinder 60 is designed to slide within the housing 40 when the head 62 is impacted by a load from the adjacent freight car or the like. The energy of the impact is absorbed by the spring assembly 50. The spring assembly 50 is positioned within the telescoping cylinder 60 between the block 44 and the head 62. A spring follower plate 52 is provided at each end of the spring assembly 50 to control the application of force to the spring assembly.

Furthermore, the spring assembly 50 comprises a plurality of stacked, hollow elastomer spring members 54. The spring members 54, eight of which are included in the illustrated buffer 30, have been formed by the application of a compressive force to a hollow elastomer body as described above with respect to the spring member 20 shown in FIGS. 1 through 3. Each spring member 54 has an expanded central core opening 56 formed by the application of an axial force to the spring in accordance with this invention. Also, a plurality of pressure plates 58 are provided in the spring assembly 50 so that one plate 58 is positioned between adjacent spring members 54. These plates 58 assist in maintaining the spring members 54 in the proper stacked alignment during the operation of the buffer 30. As explained further in U.S. Pat. No. 4,198,037, the plates 58 can be provided with surface incongruities which will form a mechanical bond with the adjacent elastomer spring members 54.

The operation of the buffer 30 is apparent from the above description of the component parts. When a force load is applied to the head 62, such as by the impact between adjacent cars in a freight train, the force energy is cushioned and absorbed by the spring assembly 50. This energy absorption results from the fact that the force telescopes the cylinder 60 inside of the housing 40 and thereby compresses the spring assembly 50 axially. When the force is relieved, the spring assembly 50 will rebound and the head 62 will return to its initial position.

FIG. 5 illustrates a modified hollow body 70 produced in accordance with the present invention. The body 70 is generally cylindrical in configuration, and is provided with a central axial bore 72. The axial ends of the cylinder 70 define an upper end 74, and a lower end 76. In this modification, each of the ends 74 and 76 is provided with a projecting neck portion 74A and 76A. As seen in FIG. 5, the necks 74A and 76A project beyond the associated end walls 74 and 76, and are axially aligned with the central bore 72. These neck portions 74A and 76A can be formed on the cylindrical body 70 by machining, or can be molded into the body if the cylinder 70 is produced by a molding operation. In the illustrated embodiment, the neck portion 74A is axially longer than the neck portion 76A. Of course, these neck dimensions can be varied to accommodate the requirements of particular applications.

In FIG. 5 the cylindrical body 70 is shown positioned between a pair of compression jaws 'J' of a suitable compression device. The jaws 'J' include recesses for receiving and confining the neck portions 74A and 76A of the body 70. The jaws 'J' are thereby arranged to apply an axial compressive force to the body 70 without deforming the neck portions 74A and 76A. In accordance with this invention, the hollow body 70 is subjected to an axial compressive force which reduces the height of the body 70 by an amount equal to at least 30% of the initial body height. As explained further above, the cylindrical body 70 is thereby formed into a compressed hollow body 70A illustrated in FIG. 6. Since the jaws 'J' include recesses for the neck portions 74A and 76A, the neck portions of the resulting compressed hollow body 70A are unchanged. However, as seen in FIG. 6, the height of the body 70A is permanently reduced compared to the body 70 shown in FIG. 5. Moreover, the axial bore 72 provided in the body 70 has been symmetrically expanded about the axial center line to form an enlarged symmetrical core portion 72A. The compressed body 70A is thereby formed to have the characteristics and advantages of the present invention, and is further provided with axial neck portions 74A and 76A which facilitate the positioning and operation of the body 70A in particular industrial applications.

FIG. 7 illustrates the utilization of the compressed hollow body 70A, such as shown in FIG. 6, in a motor mount assembly. In this typical motor mount application, a spring compression unit is positioned to dampen or cushion the vibrational energy between a relatively rigid frame member 80, and a pair of mobile motor mount members 82A and 82B. The motor mount assembly must be capable of absorbing or dampening the vibrational energy in multiple directions, such as when the motor mount members 82 vibrate both upwardly and downwardly with respect to the rigid member 80 illustrated in FIG. 7. The motor mount assembly must also be capable of absorbing at least a minimum amount of horizontal vibrational energy.

To accomplish these purposes, a pair of hollow elastomer spring bodies 70A, such as illustrated in FIG. 6, are positioned in axial alignment in the motor mount assembly illustrated in FIG. 7. The hollow configuration for the bodies 70A allows the design parameters of the body to be varied to provide a spring rate which is sufficiently low to allow the use of the copolyester polymer elastomer material in many typical motor mounting or vibration dampening operations. One of the bodies 70A is positioned between the rigid frame 80 and the motor mount plate 82A, and the other body 70A is positioned between the frame 80 and the lower motor mount plate 82B. A bolt and nut assembly 84 extends through the enlarged axial bores 72A of the bodies 70A to maintain the bodies in axial alignment.

The rigid frame 80 includes an aperture 81, and the motor mount plates 82A and 82B include apertures 83, to position the bolt assembly 84 in the motor mount assembly. As seen in FIG. 7, the apertures 81 and 83 are dimensioned to receive within a close tolerance the neck portions 74A and 76A provided on each of the bodies 70A. The bolt assembly 84 passes through the neck portions 74A and 76A. The neck portions 74A and 76A thereby secure the bodies 70A in the proper axial position with respect to the frame 80 and the motor mount plates 82A and B. The different axial sizes of the neck portions 74A and 76A accommodate the axial lengths of the apertures 81, 83 into which the neck portions extend.

The bolt assembly 84 will thereby transmit vibrational forces to the hollow body members 70A in both the upward and downward vertical directions, as illustrated in FIG. 7. Thus, if an upward force is applied to the bolt assembly 84 by the motion of the motor and the plates 82, the motor mount plate 82B is urged upwardly, as viewed in FIG. 7. The plate 82B compresses the lower member 70A against the rigid frame 80, to thereby dampen the vibrational energy caused by the motion of the motor. Likewise, a downward vibratory force applied to the bolt 84 and the plate 82A compresses the upper hollow body members 70A against the rigid frame 80, to thereby absorb the vibrational energy of the motor motion. Furthermore, the positioning of the neck portions 74A and 76A within the apertures 81 and 83 permits the bodies 70A to absorb and react to lateral or torsional loads applied by non-vertical relative motion between the plates 82A and 82B and the rigid frame 80. The hollow bodies 70A will thus dampen the vibrational energy of the motion of the motor assembly by reacting to force components in the horizontal as well as the vertical direction. The vibration damping capabilities of the assembly will thus be significantly improved.

FIG. 8 illustrates an application of the hollow bodies produced by the present invention which capitalizes upon the symmetry of the bodies. In FIG. 8 a hollow body 90, including an enlarged central core 92, is provided with neck or hub portions 94 and 96. The body 90 is similar to the hollow body 70A illustrated in FIG. 6, and can be produced to have the hub portions 94 and 96 in the same manner as described above with respect to the formation of the neck portion 74A and 76A on the hollow body 70A. The hub portions 94 and 96 on the body 90 facilitate the insertion of an axle shaft 98 into the central core 92 of the body, and through the hub portions. A flange 99 is provided on the shaft 98, as seen in FIG. 8, to bear against the hub portion 94 and restrain the wheel 90 from lateral movement to the left in FIG. 8. As also seen in FIG. 8, the right portion of the axle 98 is rotatively mounted in an appropriate aperture provided in the frame member 100. The frame 100 can be associated with a vehicle or other mobile apparatus such as a work cart or the like, on which the wheels 90 are used. A conventional retaining washer 102 can be provided to secure the axle 98 onto the frame 100 and to prevent the wheel 90 and the axle 98 from lateral movement with respect to frame 100. The symmetrical nature of the hollow bodies produced pursuant to this invention results in a generally light weight and smoothly operating wheel 90.

FIGS. 9–11 illustrate the use of the present invention in a formation of hollow bodies having shapes other than cylindrical. In FIG. 9 the original co-polyester polymer elastomer material is molded or machined to have an oval or ellipsoidal configuration. This oval elastomer body 110 is also provided with an oval central core opening 112. The core 112 is similar in shape to and co-axial with the body 110 to assure that the resulting compressed hollow body is symmetrical about its vertical axis. In the embodiment shown in FIGS. 9 and 10 the core 112 extends through the body 110.

FIG. 10 illustrates the configuration of the hollow body 110A after the above-described body 110 is subjected to the compressive force pursuant to this invention which reduces the initial axial height of the body 110 by at least 30%. The compressed hollow body 110A has an enlarged core 112A, and is provided with the enhanced spring characteristics as described above. FIG. 10 illustrates the symmetry that the body 110A and the hollow core 112A have with respect to the major axis of the oval body. The symmetry of the hollow body 110A and the central core 112A also exists in a similar fashion about the minor axis of the body.

The oval body 110A is usable as a compression spring in situations where the enlarged major axis may have some benefit, such as in a rectangular spring assembly. Of course, it will be appreciated by those skilled in the art that the shape or configuration of the hollow bodies and the internal cores can be varied to suit the particular industrial application.

FIG. 11 illustrates another modification of the invention where the oval body 110B is provided with a hollow core 112B that extends only partially through the body. The resulting body 110B has characteristics similar to the above-described body 110A, except that a solid end wall 113 is formed at one axial end of the body. This end wall 113 is useful in particular applications, such as when the body 110B is used as an energy absorption bumper. In such an application, the body 110B could be mounted on the vehicle, guard rail, or other structure in the desired position by securing a fastener, such as a bolt, to the end wall 113.

FIG. 12 illustrates the performance characteristics under dynamic and static testing conditions of a buffer 30 illustrated in FIG. 4, having the compression spring assembly 50 provided with rubber circular spring members having a central opening. The graph of FIG. 12 illustrates the travel, in millimeters, of the spring assembly 50 upon the application of a force, expressed in kilo-newtons, applied to the buffer head 62 to compress the stack of rubber springs. The solid line curve S in FIG. 12 represents the force-travel curve of the buffer when subjected to a static load applied at a standard rate of 25 millimeters of deflection per minute until full deflection occurred. The initial preload force was one kilo-newton, and the end force was 630 kilo-newtons. The area below the static compression curve S in FIG. 12 graphically illustrates the total energy ($W_e$) stored by the spring assembly 50 in a spring stroke of approximately 105 millimeters. In a typical static test, this energy ($W_e$) was nine kilo-joules for the rubber compression springs.

The force-travel curve D in FIG. 12 represents the force-distance curve generated by the application of a dynamic load to a buffer such as illustrated in FIG. 4, having the above described rubber compression springs. The dynamic load was applied by a 27,000 pound drop hammer impacting the head 62. Again, the area below the curve D in FIG. 12 graphically represents the total energy ($W_e$) stored by the rubber spring assembly through a spring stroke of approximately 105 millimeters. In a typical dynamic test for this type of buffer having rubber springs, the stored energy ($W_e$) was 18 kilo-joules. The end force was about 1325 kilo-newtons.

FIG. 13 represents graphically the same type of dynamic and static tests applied to a buffer 30 as shown in FIG. 4 which included a stack of hollow elastomer spring bodies 54 produced pursuant to the present invention. The solid line curve S in FIG. 13 represents the force-travel curve generated by the application of a static test load to the buffer 30, including the elastomer hollow bodies 54 by a screw-type loading device that deflected the spring at 25 millimeters per minute. As illustrated in FIG. 13, the maximum deflection of about 105 millimeters occurred at a force of 1050 kilo-new-

tons. The area under the curve S, representing the spring energy stored within the 105 millimeters deflection ($W_e$), was 32 kilo-joules in a typical test. In a similar manner, the broken line curve D illustrated in FIG. **13** represents the force-travel curve for the polymer spring assembly **50** under dynamic test conditions. The dynamic force was applied by the impact of a 27,000 pound drop hammer. The maximum dynamic loading of the polymer spring assembly, at a maximum deflection of 105 millimeters, was 1552 kilo-newtons. The area under the curve D, representing the energy ($W_e$) stored by the spring assembly **50**, was approximately 50 kilo-joules in a typical dynamic test.

A comparison of FIGS. **12** and **13** establishes that the co-polyester polymer springs **54** have enhanced spring characteristics and energy absorption properties when used in a buffer such as shown in FIG. **4**, as compared to rubber compression springs. Under both dynamic and static test conditions, the amount of force necessary to fully compress the spring assembly was substantially greater with the co-polyester polymer springs **54**. Moreover, the energy ($W_e$) stored by the spring, was substantially greater with the co-polyester polymer compression springs **54**. In the static tests, the energy stored increased by a factor of four from 9 to 32 kilo-joules. In the dynamic tests, the stored energy rose from 18 kilo-joules with rubber to 50 kilo-joules with the springs pursuant to this invention.

FIG. **14** illustrates additional features and characteristics of the hollow elastomer springs produced pursuant to the present invention. In FIG. **14**, four load-deflection diagrams are presented for spring members which have varying physical characteristics. In each instance the cylindrical body is made from a block of copolyester polymer elastomer material having an outside diameter of 2.5 inches and an initial axial height of 3 inches. In each instance the elastomer pad was subjected to an axial compressive force which compressed the pad by 70% of its original height. The force was then released. The pad $P_1$ was a solid elastomer cylinder. After the compression force was released pad $P_1$ had a free height of 1.61 inches. The pad $P_2$ was provided with an axial bore having an initial internal diameter of $\frac{3}{4}$ of an inch. The free height of $P_2$ after the 70% compression force was 1.64 inches. The pad $P_3$ was provided with an initial internal axial bore having an internal diameter of 1 inch. The resulting free height after compression of pad $P_3$ was 1.78 inches. Finally, the pad $P_4$ was provided with an initial axial bore having an internal diameter of $1\frac{1}{4}$ inches. The resulting free height of the pad $P_4$ was 2.18 inches.

The load-deflection curves illustrated in FIG. **14** demonstrate the versatility of the springs using this invention by depicting the different characteristics of the elastomer bodies provided with the different sized hollow cores. These load-deflection curves were generated by loading the final pads $P_1$–$P_4$ with a static load to produce a deflection of 25 millimeters per minute.

A comparison of the curves shown in FIG. **14** shows the effect of providing for and varying the diameter of a central core in the hollow elastomer body pursuant to this invention. As the core is provided and increases in size, such as illustrated by progressing from the pad $P_1$ through the pad $P_4$, the deflection for a given force substantially increases. Stated alternatively, the force needed for full deflection substantially decreases. The various pads $P_2$ through $P_4$ are thereby provided with substantially different spring rates (i.e., the ratio of load to deflection). The energy ($W_e$) stored by the springs during maximum deflection also varies. The $W_e$ for the solid pad $P_1$ was measured to be approximately 13,300 inch-pounds; for $P_2$ the $W_e$ value was about 9500 inch-pounds; for $P_3$ the value was about 8000 inch-pounds; and for $P_4$ the value was about 6900 inch-pounds.

Accordingly, the present invention allows a pad of thermoplastic elastomeric material to be easily modified to change the spring characteristics of the pad to meet the design requirements of particular industrial applications. The versatility of the invention thereby gives the designer the capability of custom-designing a spring to have a particular spring rate and/or energy storage capacity within a given space envelope.

The various embodiments of the invention are set forth above by way of example. It will be appreciated by those skilled in the art that modifications can be made to the method and apparatus of this invention without departing from the spirit and scope of the invention as set forth in the accompanying claims.

What is claimed is:

**1**. A method of producing a hollow body from a copolyester polymer elastomer material comprising the steps of:

providing a block of said copolyester polymer elastomer having a selected initial axial length and transverse shape and further having an initial core opening extending axially substantially through said block, said material and said opening being uniform in cross-sections perpendicular to a central axis, said material being such that upon being axially compressed at least 30% of said initial axial length the material will permanently retain a substantial portion of the length reduction after being compressed;

applying to said block having said cross-sections an axial force sufficient to compress said block at least 30% of its initial axial length to change the transverse shape of said block such that the configuration of said core opening is permanently expanded transversely outwardly to define sidewalls for the hollow body of a selected configuration; and

removing said axial force from said block.

**2**. A shock and vibration isolator produced by the method of claim **1**.

**3**. A method of making a compression spring unit employing the hollow body produced by the method as set forth in claim **1** and having a pair of pressure plates positioned on the axial ends of said compression springs, said method comprising the further steps of:

providing said plates with means to mechanically join said plates to said hollow body;

positioning said plates on the axial ends of said hollow body with the joining means in engagement with said body;

said axial force causing said elastomer material to flow and form a mechanical bond between said body and said plates.

**4**. A compression spring unit produced by the method of claim **1**.

**5**. A method of making a multiple unit compression spring assembly employing a plurality of hollow bodies produced by the method as set forth in claim **1** comprising the further steps of;

providing a plurality of pressure plates having means to mechanically join said plates to adjacent hollow bodies;

placing a plurality of said hollow bodies in axial alignment in a stack, with one of said plates positioned between each adjacent hollow body and the joining means on said plates in engagement with the associated hollow bodies;

said axial force causing the elastomer material comprising said hollow bodies to flow and form a mechanical bond between said bodies and the engaged pressure plates.

6. A multi-unit compression spring assembly produced by the method of claim 5.

7. A method of making a compression spring member comprising the steps of:

providing a block of thermoplastic elastomer material having a selected initial axial length and transverse shape and further having an initial core opening extending substantially through said block, the material and said opening being uniform in cross-sections perpendicular to a central axis,

said material being such that upon being axially compressed at least 30% of said initial axial length the material will permanently retain a substantial portion of the length reduction after being compressed;

applying an axial force sufficient to compress said block, having said uniform cross-sections, at least 30% of its initial axial length to change the transverse shape of said block such that the configuration of said core opening is permanently expanded transversely outwardly to thereby provide a hollow compression spring member having sidewalls of a selected configuration; and

removing said axial force from said block.

8. A method in accordance with claim 1 or claim 7 wherein the core opening extends partially through the block of elastomer material.

9. A method in accordance with claim 1 or 7 wherein the core opening extends axially through the block of elastomer material.

10. A method in accordance with claim 7 wherein said material comprises a copolyester polymer elastomer material.

11. A method in accordance with claim 1 or 7 wherein said axial force is sufficient to compress said block in the range of about 30% to 80% of its initial axial length.

12. A method in accordance with claim 11 wherein said axial force compresses said block to about 50% of its initial axial length.

13. A method in accordance with claim 1 or 7 wherein said core opening is formed by boring an opening having a selected diameter axially into said block.

14. A method in accordance with claim 1 or 7 wherein said core opening is providing by molding an axially extending opening in said block.

15. A method in accordance with claim 1 or 7 including the step of forming at one end of said block a neck portion centrally positioned with respect to said core opening and having a selected axial extent and a transverse extent reduced with respect to said transverse shape of said block.

16. A method in accordance with claim 15 including the step of forming said neck portion at both ends of the block.

17. A method in accordance with claim 1 or 7 wherein said core opening is centrally located with respect to said block to provide sidewalls which are substantially uniform and symmetrical with respect to the axis of said hollow body.

18. A method in accordance with claim 17 wherein said transverse shape of said block defines an external surface similar to and co-axial with said core opening.

19. A method in accordance with claim 18 wherein said transverse shape of said block and said core opening are circular so that said hollow body will be generally toroidal into configuration.

20. A wheel produced by the method of claim 19.

21. A compression spring produced by the method of claim 7.

22. An energy absorption bumper produced by the method of claims 1 or 7.

23. A compression spring member comprising a body of thermoplastic elastomeric material having a selected transverse configuration and axial free height and transversely expanded hollow central core opening extending substantially therethrough, the material and said opening being uniform in cross-sections perpendicular to a central axis, said material being such that upon being axially compressed at least 30% of said initial axial length the material will permanently retain a substantial portion of the length reduction after being compressed, defined by axially compressing said body having the uniform said core opening, by an extent equal to at least thirty percent of the initial height of he uncompressed elastomeric material such that the configuration of the core opening is permanently expended transversely outwardly.

24. A spring member in accordance with claim 23 wherein said material is a copolyester polymer elastomer material.

25. A spring member in accordance with claim 23 wherein said hollow core is similar to and co-axial with respect to the transverse configuration of said body so that said hollow body is generally symmetrical about its axis.

26. A spring member in accordance with claim 23 wherein said block is formed with a neck portion axially aligned with said core on at least one axial end of said block and extending axially from said end and adapted to absorb lateral and torsional forces applied to said spring.

27. A spring in accordance with claim 26 wherein a neck portion is provided on both axial ends of said block.

28. A method of producing an elastomeric compression spring having a selected spring rate comprising the steps of:

providing a block of copolyester polymer elastomer material having a selected initial axial height and transverse configuration, said material being such that upon being axially compressed at least 30% of said initial axial length the material will permanently retain a substantial portion of the length reduction after being compressed

forming a core opening in said block extending axially substantially therethrough and having a transverse size selected to define sidewalls which provide said selected spring rate, the material and said opening being uniform in cross-sections perpendicular to a central axis;

applying to said block having said core opening an axial force sufficient to compress said block at least 30% of said initial axial height such that the configuration of said core opening is permanently ex-

panded transversely outwardly to define said sidewalls; and

removing said axial force from said block to thereby provide said compression spring.

29. A method in accordance with claim 28 wherein said core is formed in said block to extend axially therethrough.

30. A method in accordance with claim 28 wherein an end wall is formed in said spring by extending said core opening only partially through said block.

31. A method of producing a spring in accordance with claim 28 including annealing said block before the application of said compressive force thereto.

32. A vibration dampening system for use as a motor mount or the like comprising:

first mounting means defining a relatively rigid frame structure;

second mounting means defining a relatively movable frame structure adapted to be connected to the source of vibration energy to be dampened; and

a pair of hollow compression spring members positioned between said first and second mounting means and arranged to be compressed to thereby dampen the vibratory motion between said mounting means;



each of said hollow spring members comprising a body of thermoplastic material having a hollow central core opening extending substantially therethrough, said material and said opening being uniform in cross-sections perpendicular to a central axis, said material being such that upon being axially compressed at least 30% of said initial axial length the material will permanently retain a substantial portion of the length reduction after being compressed, and defined by axially compressing said body with the uniform core opening of an initial configuration by an extent equal to at least 30% of the initial uncompressed height of said body to permanently set the body and to permanently change the hollow central core opening from its initial configuration to a permanent expanded configuration.

33. A system in accordance with claim 32 wherein said bodies are formed from copolyester polymer elastomeric material.

34. A system in accordance with claim 32 or 33 wherein said bodies are provided with axially extending neck portions which engage with said mounting means and dampen the lateral and torsional motion between said mounting means.

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US005316240A

# United States Patent [19]

**Girard et al.**

[11] **Patent Number: 5,316,240**

[45] **Date f Patent: May 31, 1994**

[54] **METHOD AND DEVICE FOR FILTERING THE VIBRATORY EXCITATIONS TRANSMITTED BETWEEN TWO PARTS ESPECIALLY BETWEEN THE ROTOR AND THE FUSELAGE OF A HELICOPTER**

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[21] Appl. No.: **928,306**

[22] Filed: **Aug. 12, 1992**

[30] **Foreign Application Priority Data**

Sep. 28, 1991 [FR] France ................................ 91 10722

[51] **Int. Cl.**[5] ........................ **B64C 27/51; F16F 7/00**

[52] **U.S. Cl.** ................................ **244/17.27**; 188/380; 248/550

[58] **Field of Search** ........................ 244/17.13, 17.27; 188/379, 380; 248/550

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*Attorney, Agent, or Firm*—Fisher, Christen & Sabol

[57] **ABSTRACT**

The present invention relates to a device for elastic linking between two parts (3, 5) in order to filter the vibratory excitations which are transmitted from one to the other, comprising:

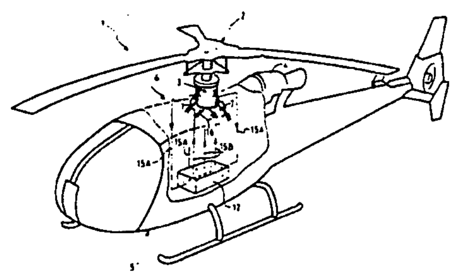
at least one linking element (7) between said parts (3, 5) comprising a transmission member (8) for the static force between said parts (3, 5), and an actuator (9) associated with said transmission member (8),

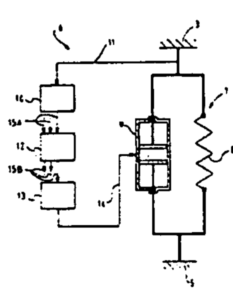
at least one means of measurement (10) of a physical quantity which is representative of said vibratory excitations, and able to supply corresponding first signals, and

electronic processing means (12) for said first signals in order to convert them into second control signals for said actuator (9).

According to the invention, said measurement means (10) is mounted on said linking element (7).

**21 Claims, 3 Drawing Sheets**





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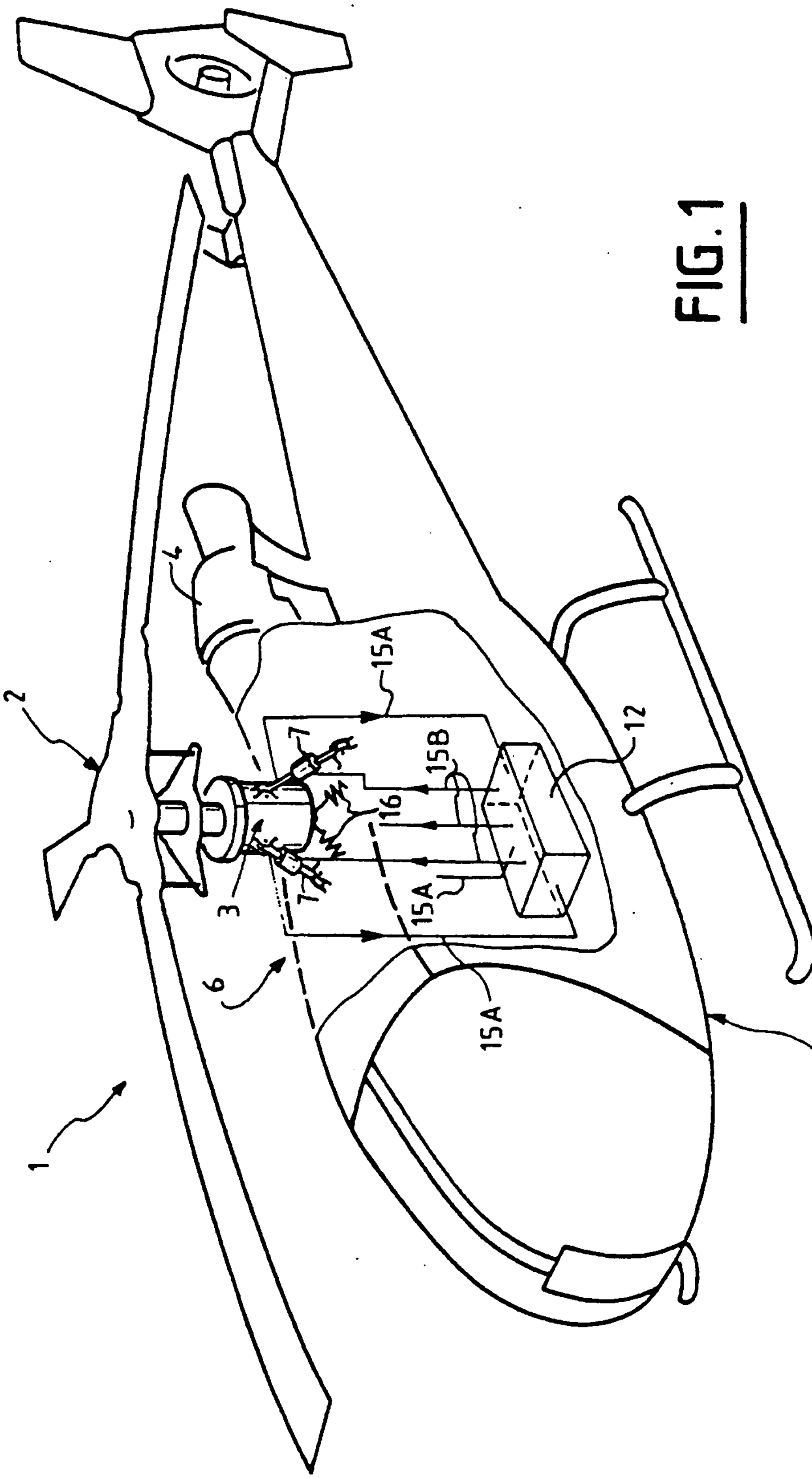


FIG.1

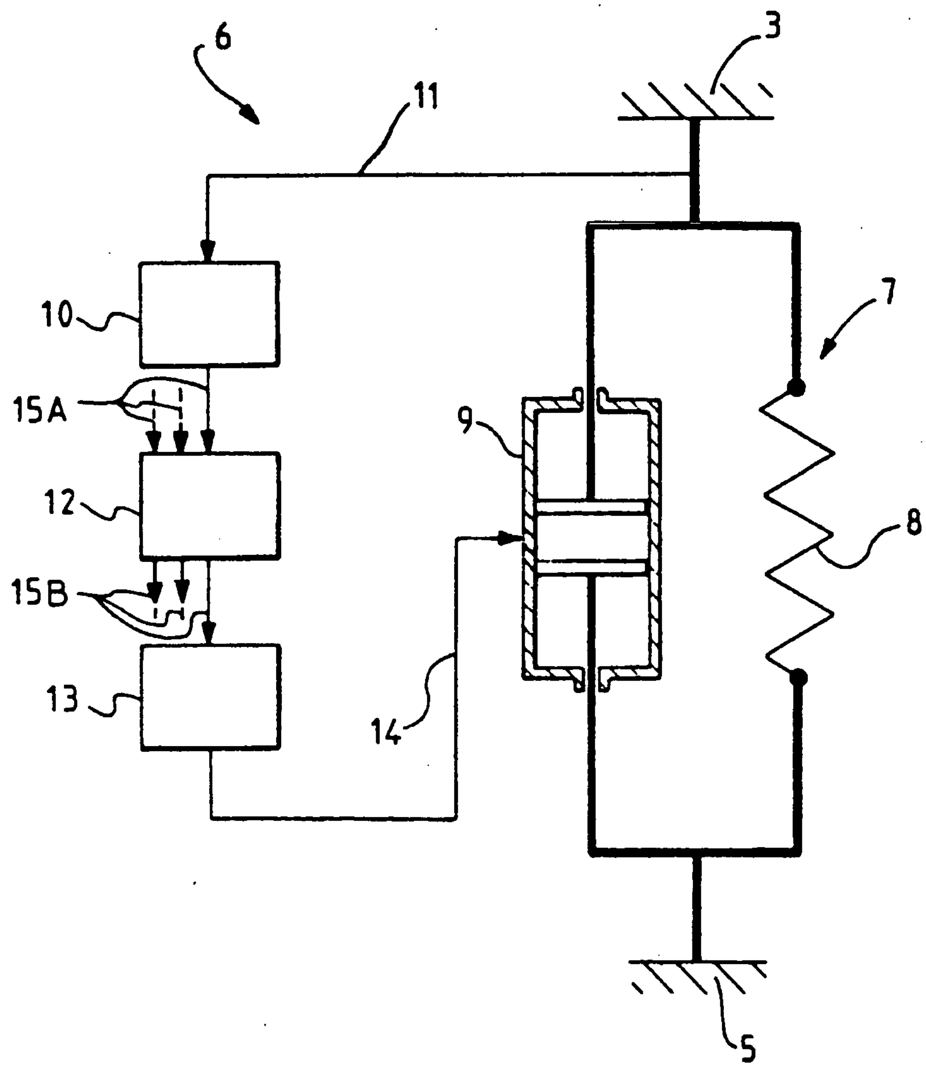


FIG. 2

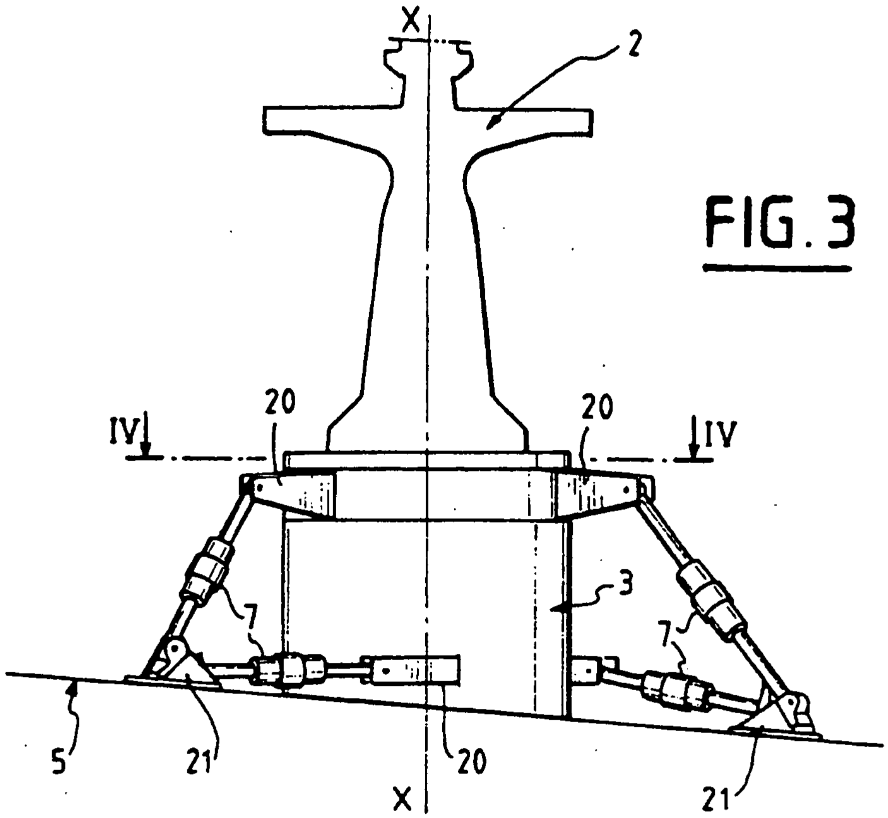


FIG. 3

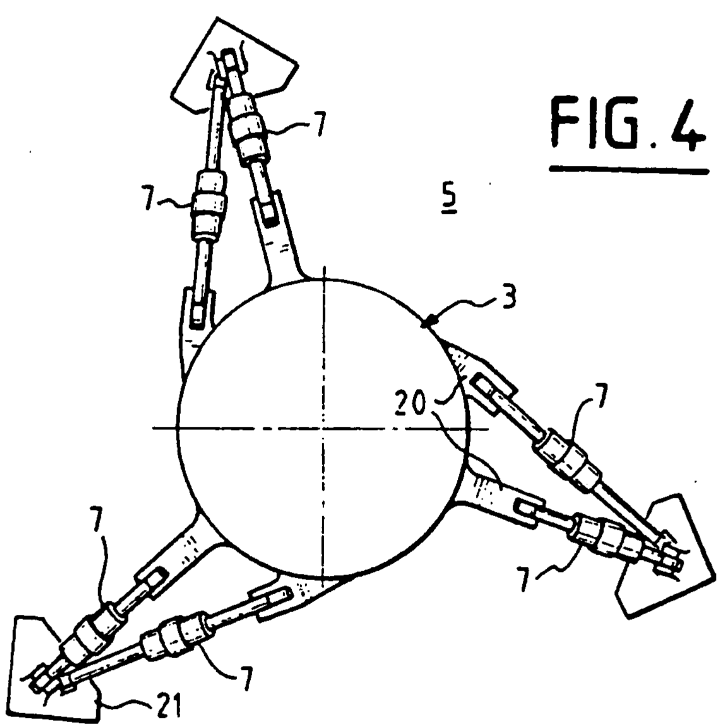


FIG. 4

# METHOD AND DEVICE FOR FILTERING THE VIBRATORY EXCITATIONS TRANSMITTED BETWEEN TWO PARTS ESPECIALLY BETWEEN THE ROTOR AND THE FUSELAGE OF A HELICOPTER

## BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a method for filtering the vibratory excitations transmitted between two parts and an elastic linking device between two parts which transmits the static forces from one to the other in the axis of the device and for simultaneously filtering the associated coaxial vibratory excitations which are transmitted from one to the other.

More particularly, although not exclusively, such a device can be used in the suspension linking the main transmission gearbox to the fuselage of an aircraft with rotating wings, such as a helicopter, in order to filter the vibrations generated by the rotor and transmitted to the fuselage of said aircraft by said transmission gearbox.

2. Background Art

In fact, one of the fundamental problems of the helicopter arises from the general vibratory level which conditions, on the one hand, the level of the alternating stresses throughout the machine (and consequently the fatigue behavior and hence the lifetime of the parts) and, on the other hand, the cabin comfort and control vibrations.

The object of much research has therefore been to attenuate, if not to completely cancel out, this vibratory level inherent in the very operation of the rotor which not only transmits to the hub static or quasistatic forces and moments created by:

- the lift (perpendicular to the plane of the rotor),
- the drag (in the plane of the rotor and parallel to the component $V_H$, normal to the rotor mast, of the forward speed of the aircraft),
- the drift force (perpendicular to the preceding two and also in the plane of the rotor) which remains small and which can generally be ignored, but also periodic forces and moments originating from aerodynamic dissymetries (lift and profile drag) which appear during the rotation of the blades, essentially due to the forward speed in flight of translation at high speed, or also dissymetries resulting from the inequality of distribution of the speeds induced on the disk of the rotor at low speed (transition area). These alternate aerodynamic forces and moments are transmitted to the center of the rotor after having been attenuated or amplified by the blades.

Given that, in a general way, by $\Omega$ the speed of rotation of the rotor is expressed in number of revolutions per second and by b represents the number of blades, note that:

- the forces (due to the flapping movements of the blades) and the moments (due to the drag movements of the blades) whose axes are carried by the axis of the rotor, are transmitted to the mast and to the fuselage only if their frequency expressed in hertz (Hz) is a harmonic of $b\Omega$, and thus of the form $kb\Omega$ (k : positive integer, equal to or greater than 1. The transfer of these forces and.. moments from the rotating axes to the fixed axes takes place without frequency change (oscillation and torsion effect in the structure);
- the forces (due to the drag movements of the blades) and the moments (due to the flapping movements of the blades) whose axes are in the plane of the rotor, are transmitted to the mast and to the fuselage only if their frequency is of the form $(kb\pm1)\Omega$, the resulting forces and moments then being at the frequency $kb\Omega$ in fixed axes (roll and pitch effects, transverse or longitudinal sway, principally in $b\Omega$).

Consequently, note that a balanced rotor transmits, over and above the static forces and moments, only alternate forces and moments at a frequency which is a multiple of the speed of the rotor multiplied by the number of blades, the fundamental frequency being equal to $b\Omega$.

It would thus be appropriate, in order to avoid dangerous periodic forces at a frequency which is a multiple of the speed of the rotor, to increase the number of blades since:

- the excitation harmonics at the site of the blades which affect the vibrations in the fuselage are distributed according to the order below:

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| two-blade | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 |
| three-blade | | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 |
| four-blade | | | 3 | 4 | 5 | | 7 | 8 | 9 | | 11 |
| five-blade | | | | 4 | 5 | 6 | | | 9 | 10 | 11 |

- the higher the order of the harmonics, the smaller their amplitude;
- the excitations which affect comfort are the harmonics $(kb\pm1)\Omega$ in the axes of the blades found, after composition, at frequencies $kb\Omega$ in the fuselage of the helicopter;
- the lower the excitation frequency, the greater the extent to which people are sensitive to it, especially along the vertical axis.

For reasons cost and of complexity, it is nevertheless appropriate to limit the number of blades.

Moreover, it is well known that as performance rises, the excitations increase as $V^n$ $(n>1)$: at high speed, the vibratory level in the fuselage grows in the same way.

These comments, together with evidence for ever more important comfort imperatives, are the justification for devising systems capable of transmitting the static forces and moments designated by $F_0$ while attenuating the vibrations, which correspond to the decomposition into a Fourier series $\Sigma Fn \cos n\Omega t$ (t: time, n: order of the harmonics). This attenuation must especially tend to minimize the vertical components of the dynamic loading at the level of the fuselage, which turn out to be the most troublesome in practice.

The parameters which condition the vibratory levels are accounted for of at the design stage of a rotor so as to minimize the effects thereof: type of hub (rigid or articulated hub) and choice of the number of blades, aerodynamic optimization of the blades in order to reduce the excitations and optimization of the dynamic response of the blades in order to reduce the torque vector (forces and moments) transmitted to the head of the rotor.

When these choices and the compromises that are reached do not yield the theoretical (or. experimental) results anticipated for the vibratory levels, complementary means of action are resorted to in order to modify the excitation torque vector applied to the fuselage:

1) at the level of the rotor, with adoption of passive pendular antivibrators (or mass-spring systems ar-

ranged on the head of the rotor, for example) or of an active multi-cyclical control. In this latter case, a computer transmits signals to the cyclic pitch control of the blades by means of servocontrols, analyzes the effect produced and optimizes the input so as to have minimal acceleration as output. In other words, the multi-cyclical control is a solution specific to helicopters and based on the modification of the aerodynamic forces applied to the blades (and, consequently, on the modification of the excitation torque vector at the head of the rotor, thus at the exterior of the fuselage) by multicyclical injection of the pitch commands;

2) at the level of the fuselage (designed at the outset with, among other imperatives, that of not having a characteristic vibration mode too close to the main excitation frequencies) by:

local processing, by modifying the shapes of the dynamic responses of the structure (batteries moved, stiffening of elements) or by installing mass-spring resonators (sprung batteries for example),

overall processing, by active control of the structure, based on the modification of the excitation torque vector applied to the fuselage (that is to say the distribution of the interior forces) and of the response of the latter (French Patent Nos. 1,506,385 and 2,566,862); or

3) intervention at the site of the interface between the mechanical assemblies of the main rotor and the fuselage in order to filter the transfer of the vibrations of the rotor to the airframe, especially via passive suspensions such as those described, for example, French Patent Nos. 1,507,306, 2,499,505 and 2,629,545.

More precisely, patent French Patent No. 1,506,385 relates to an attenuation method and an electrohydraulic attenuator for an aircraft with rotating wings. The method consists of creating, on the basis of the dynamic accelerations measured on the fuselage, electrical signals converted into variations in hydraulic pressure by means of an electrohydraulic servo valve, which pressure variations are transmitted to a double-acting jack arranged between the fuselage and the main transmission gearbox, in such a way as to oppose the vibrations. In order to do this, an accelerometer situated in the fuselage of the aircraft is linked to the control circuit of the double-acting jack. The jack constitutes a fourth linking bar, or, as an alternative, one of the usual bars which comprises, in this case, an elastic member in parallel which provides the flexibility necessary for the correct operation of the device while being able to take up the (substantially static) lift and maneuver forces.

A development of this concept is described in French Patent No. 2,566,862, in which, between the fuselage and the rotor of a helicopter, a plurality of actuators are provided whose oscillations are controlled in phase and in amplitude by virtue of the processing of signals which are representative of the dynamic accelerations, measured by a plurality of accelerometers arranged on the fuselage. Such a system operates in a closed loop. On the basis of the accelerometric measurements on the fuselage, the optimum commands to be generated are obtained with the use of a computer. The effective application of these commands modifies the condition of the fuselage and thus the subsequent measurements.

However, such accelerometric measurements are likely to be affected by errors and uncertainties, due especially to possible phase offsets, by the very fact that they are carried out on the fuselage of the aircraft.

## BROAD DESCRIPTION OF THE INVENTION

The object of the present invention is to avoid this drawback.

To this end, the method for filtering the vibratory tory excitations transmitted between two parts, linked by at least one linking element which comprises a transmission member for the static force between said parts, and an actuator associated with said transmission member, in which method:

a physical quantity which is representative of the vibratory excitations transmitted from one part to the other is measured, and corresponding first signals are generated, and

said first signals are processed in order to convert them into second control signals for said actuator, which is controlled in order to oppose said vibratory excitations,

is noteworthy, according to the invention, in that said physical quantity is measured on said linking element.

Hence, carrying out the measurement of said physical quantity directly on the linking element makes it possible to get round errors or uncertainties linked to measurements carried out on one of the parts in question, especially the fuselage of a helicopter.

Advantageously, as the physical quantity, the axial deformation of said linking element is measured.

Preferably, in the case where a plurality of linking elements link said parts, said physical quantity is measured on each of said linking elements.

According to another characteristic of the invention, during the processing of said first signals, the harmonic component or components of the vibratory excitations which the respective actuator has to oppose is/are identified in each linking element.

In particular, for said identification, it is possible to carry out either a time analysis in real time, by digital or analog bandpass filtering, or a Fourier analysis.

Moreover, said second control signals for the various actuators can be generated either independently or dependently.

According to another characteristic of the invention, in the case where additional passive links are provided between said parts, preferably:

said physical quantity is moreover measured on one or more of the additional passive links; and

the dependent control signals for each of said actuators are derived by applying an automatic and continuous minimization of the performance criterion PI of formula:

$$PI = \sum_{k=1}^{p} \left[ \sum_{i=1}^{N+M} a_i(\epsilon_{kf})_i^2 + \sum_{(i \neq j)=1}^{N+M} a_{ij}(\epsilon_{kf})_i(\epsilon_{kf})_j \right]$$

in which:

N = number of actuators and corresponding measurements,

M = number of measurements on the additional passive link or links,

$\epsilon_{kf}$ = harmonic component of rank k of said physical quantity of fundamental frequency f,

p = number of harmonic components to be filtered,

$|a|$ = weighting matrix for the effect of each linking element.

The present invention also relates to an elastic linking device between two parts which transmits, from one to the other, the static forces in the axis of the device, and simultaneously filters the associated coaxial vibratory excitations which are transmitted from one to the other, comprised of:

at least one linking element between said parts which comprises a transmission member for the static force between said parts and an actuator associated with said transmission member;

at least one means of measurement of a physical quantity which is representative of the vibratory excitations transmitted from one part to the other, and able to supply corresponding first signals; and

electronic processing means for said first signals in order to convert them into second control signals for said actuator, which is controlled in order to oppose said vibratory excitations.

Said device being noteworthy in that said measurement means is mounted on said linking element.

In the case where the device comprises a plurality of said linking elements between said parts and a plurality of said measurement means, a measurement means is, advantageously, mounted on each of said linking elements.

Preferably, each measurement means is a sensor which measures the axial deformation of the respective linking element, especially an extensometric gauge.

Moreover, said actuator may be a double-acting jack mounted in parallel on said linking member, and controlled by means of a solenoid valve.

Advantageously, said electronic processing means comprise analysis means for identifying, in each linking element, the harmonic component or components of the vibratory excitations which the respective actuator has to oppose. In particular, said analysis means may carry out a time analysis in real time by digital or analogue bandpass filtering, or a Fourier analysis.

Moreover, the device in which additional passive links are provided between said parts is noteworthy in that, preferably, additional measurement means for said physical quantity are moreover mounted on said additional passive link or links and the dependent control signals for each of said actuators are derived by applying an automatic and continuous minimization of the performance criterion PI of formula:

$$PI = \sum_{k=1}^{p} \left[ \sum_{i=1}^{N+M} a_i(\epsilon_{kf})_i^2 + \sum_{(i \neq j)=1}^{N+M} a_{ij}(\epsilon_{kf})_i(\epsilon_{kf})_j \right]$$

which:

$N$=number of actuators and corresponding measurements,

$M$=number of measurements on the additional passive link or links,

$\epsilon_{kf}$=harmonic component or rank k of said physical quantity of fundamental frequency f,

$p$=number of harmonic components to be filtered,

$|a|$=weighting matrix for the effect of each linking element.

The figures of the attached drawing will clarify how the invention can be produced.

## BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic view in perspective of a helicopter which incorporates the elastic linking device according to the invention.

FIG. 2 schematically illustrates an elastic linking device according to the invention.

FIG. 3 shows an example of arrangement of the linking elements of the device of the invention between the main transmission gearbox and the fuselage of a helicopter.

FIG. 4 is the section along line IV—IV of FIG. 3.

## DETAILED DESCRIPTION OF THE INVENTION

As seen in FIG. 1, the helicopter 1 comprises a main lift rotor 2 driven by a main transmission gearbox 3 for the motive power provided by the engine 4, the gearbox supporting the fuselage 5.

Between the main transmission gearbox 3 and the fuselage 5 an elastic linking device 6 is provided, which comprises a plurality of linking elements 7 (whose number is equal to three at least), linked, on the one hand, to the main transmission gearbox 3 and, on the, other hand, to the fuselage 4 and each which comprises, as shown in FIG. 2, a transmission member 8 for the static force between the main transmission gearbox 3 and the fuselage 5 (with stiffness adapted to optimal operation of the device), and an actuator 9 associated in parallel with the transmission member 8, and which may be, as illustrated, a double-acting jack which could generally be as described in FR-1,506,385. A sensor 10 which is mounted on each linking element 7, by being integrated or linked in any appropriate way to the latter (as is illustrated very schematically in FIG. 2 by the link 11), makes it possible to measure a physical quantity which is representative of the vibratory excitations transmitted from the main transmission gearbox 3 to the fuselage 5. Such a sensor, which may be an extensometric gauge measuring the axial deformation of the respective linking element (especially the transmission member 8 for the static force), supplies first electrical signals (via the link 15A) to a computer 12 which converts said first signals into second control signals for the actuator 9 (link 15B), controlled by means of the solenoid valve 13 (link 14), in order to oppose the vibratory excitations (the source of power for the actuator is not represented).

These second control signals may be generated independently or dependently, according to whether or not the control of a given actuator is dependent on the control of the other actuators. Moreover, the computer 12 may be common to all the linking elements 7, as is shown symbolically by the additional inputs 15A and outputs 15B of the computer 12 in FIGS. 1 and 2.

As explained in more detail below, the computer 12 comprises analysis means for identifying, in each linking element 7, the harmonic component or components of the vibratory excitations which the respective actuator 9 has to oppose, by carrying out either a time analysis in real time, by digital or analogue bandpass filtering, or a Fourier analysis.

The device of the invention therefore uses modules 7 which are constituted by the parallel association, on the one hand, of a member 8 with optimum passive stiffness for transmitting the static forces (lift of the aircraft) or quasistatic forces (slow maneuvers) and, on the other hand, of an actuator 9, driven in force, in order to mini-

mize, or even cancel out, the excitation harmonics at frequencies $kb\Omega$ ($k=\text{integer}\geqq 1$; $b$=number of blades; $\Omega$=speed of rotation of the rotor in revolutions per second, frequencies in Hz) transmitted from the head of the rotor to the fuselage.

It is obvious that, in order to filter all the harmonics (multiples of the rotor rotation speed multiplied by the number of blades), the command x must be of the type:

$$x = \sum_{k=1}^{y} x_{kb\Omega}$$

it being understood that the fundamental excitation and the associated command correspond to $k=1$, and that y is an integer tending to $\infty$.

Hence it will be observed that, in the present invention, the closed loop is, as it were, localized in each module, while possibly taking account of the operation of the other active modules and, possibly, other passive links, as will be seen in detail below. As already indicated, this makes it possible to avoid the errors and uncertainties linked to measurements carried out on the fuselage of the aircraft.

As already mentioned, each sensor **10** may be constituted by an extensometric gauge on each active module **7** and, possibly, at the site of the interface between any additional passive linking element **16** (FIG. **1**) and the fuselage **5**, for detecting the relative deformations of the materials which are statically and dynamically loaded. The corresponding electrical signal is then processed by the computer **12** in order to control each actuator **9**. Consequently, a relative deformation $\epsilon=\Delta 1/1$ is detected, whose harmonic components $\epsilon_{kb\Omega}$[or $\epsilon_{kf}$ where f (fundamental frequency of the excitation) $=b\Omega$ ]are sought, which are representative of the harmonic components of rank k of the dynamic excitation force at the frequency $b\Omega$ to be filtered in whole or in part by each of the actuators.

Two techniques can be envisaged for deducing the components $\epsilon_{kf}$ of the overall measurement $\epsilon=\epsilon_{(t)}$; with $t$=time.

1) The Fourier analysis which consists of developing the time function $\epsilon$, which is periodic and with angular frequency $\omega$ with $\omega=2\pi b\Omega$ into a series of sinusoids of angular frequency $k\omega$ ($1\leqq k\leqq y$) and of phase angle $\phi_k$, i.e.:

$$\epsilon_{(t)}=\epsilon_0+\epsilon_{1f}\sin(\omega t+\phi_1)+\ldots+\epsilon_{kf}\sin(k\omega t+\phi_k)+$$

then in analytically determining the quantities $\epsilon_{kf}$ and $\phi_k$ ($k=1$ to y), and in adopting, for example, only "p" values of k.

The application of this method can be envisaged only if the commands are dependent, that is to say if the command for each actuator takes account of the effects of the other actuators or even of the set of measurements (according, for example, to the performance criteria proposed below). In the opposite case, the convergence of the calculation algorithms for each module is too slow, or even impossible, with respect to the actions of the other modules and, consequently, with respect to the change in the vibratory level to be controlled.

2) Time analysis which consists of filtering, by digital or analogue means with a bandpass filter and a short sampling time, the harmonic components of $\epsilon_{(t)}$ which it is appropriate to attenuate, not to say cancel out. A real-time computer (that is to say with short calculating time with respect to the preponderant time constant of the system) then derives, in real time, the electrical signals to be transmitted to the actuator. Considering these characteristics, this method applies in every case.

In the case of a single actuator controlled on the basis of the measurement of the deformation of the linking element with which it is linked or integrated, the dynamic force may be simply cancelled out. An identical result is obtained in the case of a plurality of actuators whose commands are independent and derived, for each of them, from the force exerted on the transmission support platform. The same may be even be true when the commands are dependent, as it can be demonstrated that, if a structure is controlled by N actuators, it is possible to cancel out N chosen measurements and, consequently, those relating to the dynamic elongations of the linking elements equipped with an actuator. This is an ideal theoretical solution in the absence of any passive link.

However, if the fuselage is sensitive to the excitations transmitted by the passive links arranged, for example, at the bottom of the main transmission gearbox (longitudinal or transverse sway movements, for example), the device as described above may prove to be insufficient.

In order to avoid this drawback, the first solution consists of in replacing the links between transmission gearbox bottom and fuselage by active isolator elements which are constituted by modules identical to those previously described.

As the movement of a rigid body is described by three displacements and three rotations, there are six degrees of freedom to be actively controlled. Hence, the minimal solution, which makes it possible to theoretically eliminate all transfer of dynamic excitations from the rotor to the fuselage, consists of providing for the installation of six active modules according to the invention, in accordance with the embodiment example illustrated by FIGS. **3** and **4**. This is, once again, a theoretically ideal configuration.

As can be seen in these figures, three linking elements **7** are arranged between the upper part of the main transmission gearbox **3** and the upper structure of the fuselage **5** along the edges of a trihedron whose peak is situated on the axis X—X of the main lift rotor **2**, and three other linking elements **7** are arranged between the substantially circular bottom of the main transmission gearbox **3** and the upper fuselage structure **5**, tangentially to the bottom of the main transmission gearbox and equidistant from each other. The linking elements **7** are fixed to the main transmission gearbox **3** and to the fuselage **5** by means of conventional clevises **20**, **21**.

With the assumption that the links **16** between the main transmission gearbox bottom and the fuselage remain passive, a second solution amounts to taking account of the measurement of the deformations introduced, in the fuselage, by these links. For example, if N active modules are available, linked to N deformation measurements, and P passive links **16** (for example three in number) are available, linked to M measurements, the dependent commands for the N modules can be optimized by applying, for example, the quadratic criterion according to PI, or performance criterion, to be minimized in order to establish the optimum commands to be generated:

$$PI = \sum_{k=1}^{p} \left[ \sum_{i=1}^{N+M} a_{i}(\epsilon_{kf})_i^2 + \sum_{(i \neq j)=1}^{N+M} a_{ij}(\epsilon_{kf})(\epsilon_{kf})_j \right]$$

in which $\epsilon_{kf}$ designates the harmonic component of rank k, f being the fundamental frequency bΩ, p designates the number of harmonics which it is desired to attenuate, and |a| a weighting matrix which modulates the commands to the various N actuators, thereby permitting optimal adjustment of the suspension. The dynamic forces are then non-zero in the links between the main transmission gearbox and the fuselage.

More precisely, this performance criterion is equivalent to:

$$PI = \epsilon^H a\, \epsilon$$

where $\epsilon$ is the vector of the measured relative deformation, $\epsilon^H$ the conjugate transposed matrix deduced from $\epsilon$, and |a| the weighting matrix associated with the measurements.

It will be noted that, having regard to the degrees of freedom of the passive links **16**, the number of measurements M may be different from the number P of passive links. If the measurements M relate to the displacements, for example, of the P passive links at the level of the fuselage, it turns out that, for each link P, there are three possible measurements, i.e. $M = 3 \times P$, and, for that reason, as many sensors may be provided as there are measurements to be carried out.

We claim:

**1**. A method for filtering the vibratory excitations transmitted between two parts **(3,5)**, linked by at least one linking element **(7)** comprising a member **(8)** for transmission of the static force between said parts **(3,5)**, and an actuator **(9)** associated with said transmission member **(8)**, comprising:

generating a physical quantity which is representative of the vibratory excitations transmitted from one part to the other part is measured on said linking element **(7)**, and corresponding first signals are generated;

processing said first signals by processing means **(12)**, in order to convert said first signals into second control signals for said actuator **(9)**; and

carrying out the decoupling of the static force and of the vibratory excitations by said processing means **(12)**, in such a way that said second control signals drive the actuator in order to exactly oppose said vibratory excitations.

**2**. The method as claimed in claim **1** wherein, as a physical quantity, the axial deformation of said linking element **(7)** is measured.

**3**. The method as claimed in claim **1**, in the case where a plurality of linking elements **(7)** link said parts **(3,5)**, wherein said physical quantity is measured on each of said linking elements **(7)**.

**4**. The method as claimed in claim **1** wherein, during the processing of said first signals, the harmonic components of the vibratory excitations which the respective actuator **(9)** has to oppose are identified in each linking element **(7)**.

**5**. The method as claimed in claim **4** wherein, for said identification, a time analysis is carried out in real time, for said identification, by digital or analogue bandpass filtering.

**6**. The method as claimed in claim **4**, wherein, for said identification, a Fourier analysis is carried out.

**7**. The method as claimed in claim **3** wherein said second control signals for the various actuators **(9)** are generated independently.

**8**. The method as claimed in claim **3** wherein said second control signals for the various actuators **(9)** are generated dependently.

**9**. The method as claimed in claim **3** wherein, the case where additional passive links **(16)** are provided between said parts **(3,5)**:

said physical quantity is moreover measured on each of the additional passive links **(16)**; and

the dependent control signals for each of said actuators **(9)** are derived by applying an automatic and continuous minimization of the performance criterion PI of formula:

$$PI = \sum_{n=1}^{p} \left[ \sum_{i=1}^{N+M} a_{i}(\epsilon_{kf})_i^2 + \sum_{(i \neq j)=1}^{N+M} a_{ij}(\epsilon_{kf})(\epsilon_{kf})_j \right]$$

in which:

N=number of actuators and corresponding measurements,

M=number of measurements on the additional passive links,

$\epsilon_{kf}$=harmonic component of said physical quantity,

p=number of harmonic components to be filtered,

|a|=weighting matrix for the effect of each linking element.

**10**. An elastic linking device between two parts **(3,5)** for transmitting, from one part to the other part, the static forces in the axis of the device and simultaneously filtering the associated coaxial vibratory excitation which are transmitted from one part to the other part, said device comprising:

at least one linking element **(7)** between said parts **(3,5)** comprising a transmission member **(8)** for the static force between said parts **(3,5)**, and an actuator **(9)** associated with said transmission member **(8)**;

at least one means of measurement **(10)** of a physical quantity which is representative of the vibratory excitations transmitted from one part to the other part, said measurement means being able to supply corresponding first signals, said measurement means **(10)** being mounted on said linking element **(7)**; and

electronic processing means **(12)** for processing said first signals in order to convert them into second control signals for said actuator **(9)**, said processing means **(12)** decoupling the static force and the vibratory excitations, in such a way that said second control signals drive the actuator in order to exactly oppose said vibratory excitations.

**11**. The device as claimed in claim **10** comprising a plurality of said linking elements **(7)** between said parts **(3,5)** and a plurality of said measurement means **(10)**, wherein a measurement means **(10)** is mounted on each of said linking elements **(7)**.

**12**. The device as claimed in claim **10** wherein each measurement means is a sensor **(10)** measuring the axial deformation of the respective linking element **(7)**.

**13**. The device as claimed in claim **12** wherein said sensor **(10)** is an extensometric gauge.

14. The device as claimed in claim 10 wherein said actuator is a double-acting jack (9) mounted in parallel on said linking member (8), and controlled by means of a solenoid valve (14).

15. The device as claimed in claim 10 wherein said electronic processing means (12) comprises analysis means for identifying, in each linking element (7), the harmonic component or components of the vibratory excitations which the respective actuator (9) has to oppose.

16. The method as claimed in claim 15 wherein said analysis means carries out a time analysis in real time, by digital or analogue bandpass filtering.

17. The method as claimed in claim 15 wherein said analysis means carry out a Fourier analysis.

18. The device as claimed in claim 10 wherein, in which additional passive links (16) are provided between said parts (3,5), additional measurement means for said physical quantity is mounted on said additional passive links (16) and wherein the dependent control signals for each of said actuators (9) are derived by applying an automatic and continuous minimization of the performance criterion PI of formula:

$$PI = \sum_{n=1}^{p}\left[\sum_{i=1}^{N+M} a_{ii}(\epsilon_{kf})_i^2 + \sum_{(i\neq j)=1}^{N+M} a_{ij}(\epsilon_{kf})_i(\epsilon_{kf})_j\right]$$

in which:

N=number of actuators and corresponding measurements,

M=number of measurements on the additional passive links,

$\epsilon_{kf}$=harmonic component of said physical quantity,

p=number of harmonic components to be filtered,

|a|=weighting matrix for the effect of each linking element.

19. An aircraft with rotating wings, especially a helicopter, comprising a main lift rotor (2) driven by a main transmission gearbox (3) for the motive power, which gearbox supports a fuselage (5), said aircraft comprising:

an elastic linking device between said transmission gearbox (3) and said fuselage (5) for transmitting, from said transmission gearbox (3) to the fuselage (5), the static forces in the axis of the device and simultaneously filtering the associated coaxial vibratory excitations which are transmitted from the transmission gearbox (3) to the fuselage, said device comprising; at least one linking element (7) between the transmission box (3) and the fuselage (5) comprising a transmission member (8) for the static force between said transmission box (3) and said fuselage (5) and an actuator (9) associated with the transmission member (8);

at least one means of measurement (10) of a physical quantity which is representative of the vibratory excitations transmitted from the transmission box (3) to the fuselage (5), said measurement means able to supply corresponding first signals, the measurement means (10) being mounted on said linking element (7) and

electronic processing means (12) for processing said first signals in order to convert them into second control signals for said actuator (9), said processing means (12) decoupling the static force and the vibratory excitations, in such a way that said second control signal drive the actuator in order to exactly oppose said vibratory excitations.

20. The aircraft with rotating wings as claimed in claim 19 comprising at least three said linking elements (7) arranged between the upper part of said main transmission gearbox (3) and the upper structure of said fuselage (5) along the edges of a trihedron whose peak is situated on the longitudinal axis (X—X) of the main lift rotor(2).

21. The aircraft with rotating wings as claimed in claim 20 wherein at least three other linking elements (7) are arranged between the bottom of said main transmission gearbox (3) and the upper structure of said fuselage (5), tangentially and equidistantly to the bottom of said main transmission gearbox (3).

* * * * *

# United States Patent [19]

**Mouille et al.**

[11] **Patent Number: 4,458,862**

[45] **Date of Patent: Jul. 10, 1984**

[54] **ANTIRESONANT SUSPENSION DEVICE FOR HELICOPTER**

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[21] Appl. No.: **470,704**

[22] Filed: **Feb. 28, 1983**

**Related U.S. Application Data**

[63] Continuation of Ser. No. 229,749, Jan. 29, 1981, abandoned.

[30] **Foreign Application Priority Data**

Feb. 5, 1980 [FR] France .............................. 80 02505

[51] **Int. Cl.³** .............................................. **B64C 27/06**

[52] **U.S. Cl.** ................................ **244/17.27**; 188/379; 248/557; 248/559; 416/500

[58] **Field of Search** .......................... 244/17.25, 17.27; 416/145, 500; 248/554, 556, 557, 559; 188/378, 188/379, 380

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[57] **ABSTRACT**

The invention relates to aeronautics in general, and in particular to a suspension device for helicopter, comprising a flexible mounting plate at the center of which is fixed the bottom of the main gear box whose top is supported by hinged oblique bars. The mounting plate offers radial arms hinged to the base of the bars and to the fuselage of the helicopter. These arms bear flapping weights creating forces of inertia with reactions at the attachment points of direction opposite the elastic reactions of deformation of the mounting plate. A device of this type enables the vibrations on board helicopters to be reduced.

**12 Claims, 13 Drawing Figures**

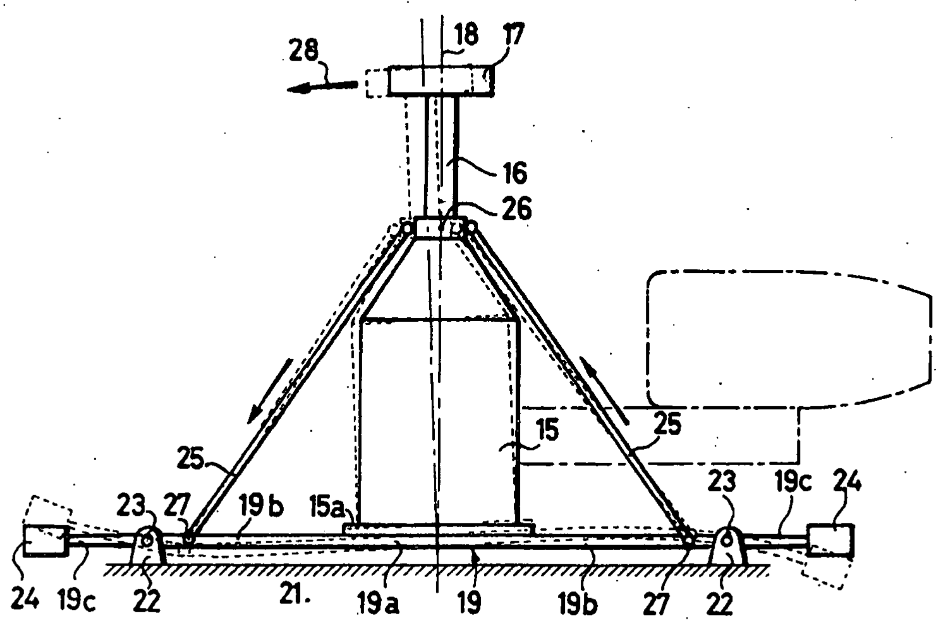


FIG.1

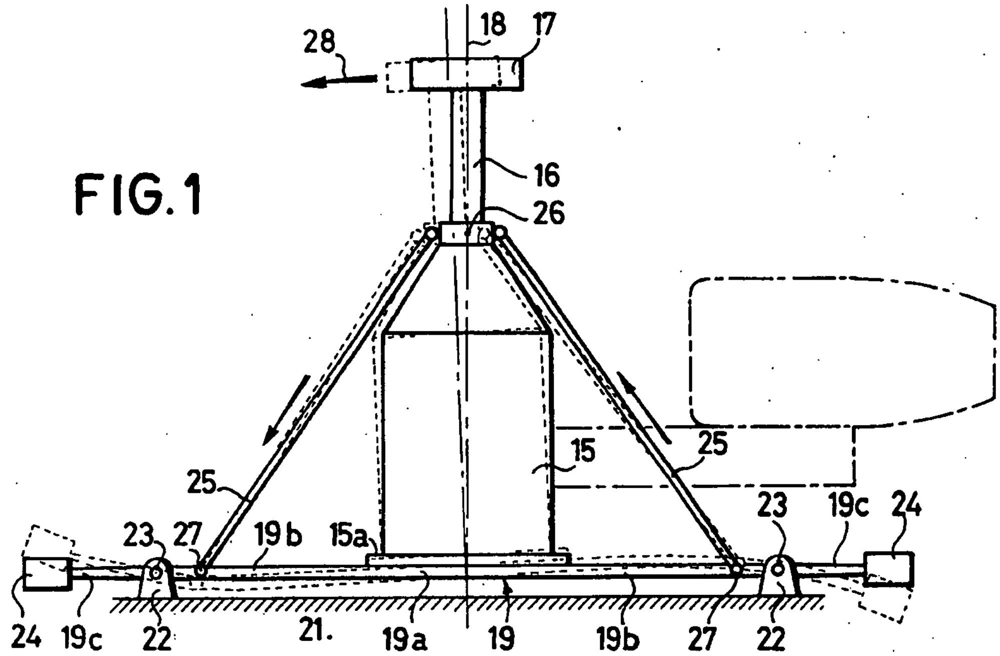
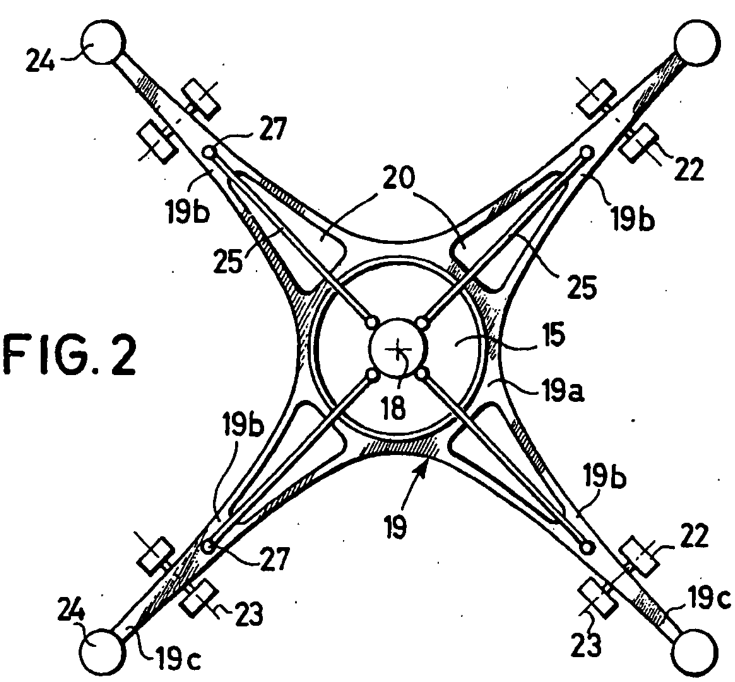

28
18
17
16
26
25
15
25
23
19c
24
23
27
19b
15a
24
19c
22
21.
19a
19
19b
27
22

FIG.2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27
20
22
19b
19b
25
25
15
19a
18
19b
19b
19
22
27
19c
23
24
23
19c

FIG.3

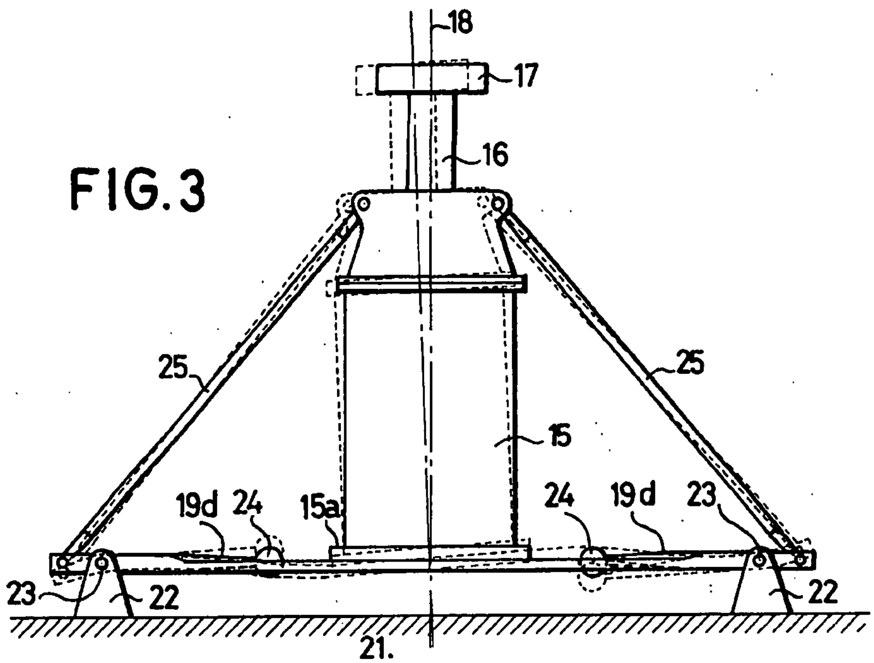
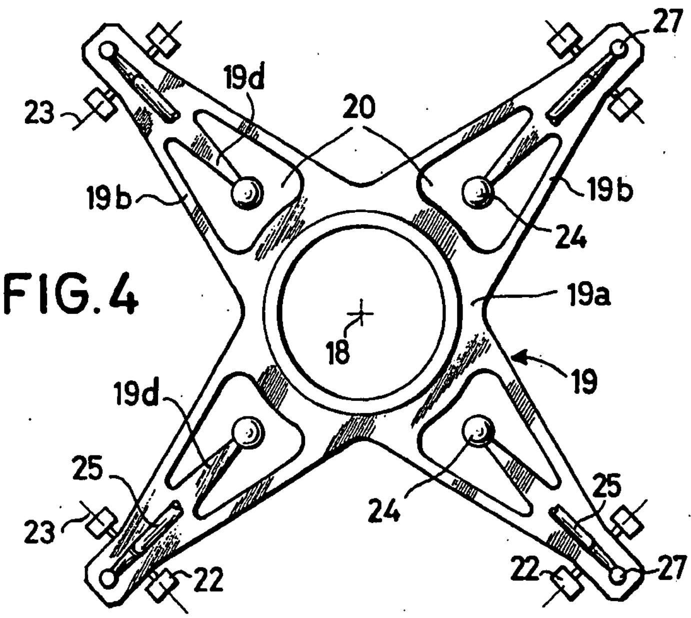

18
17
16
25
25
15
19d
24
15a
24
19d
23
23
22
22
21.

FIG.4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19d
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23
19b
19b
24
19a
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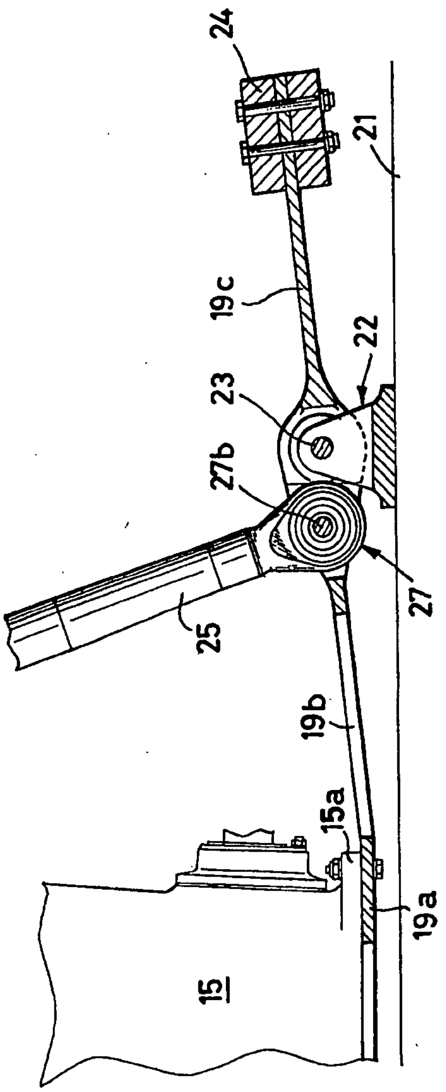
19
19d
25
25
24
23
22
22
27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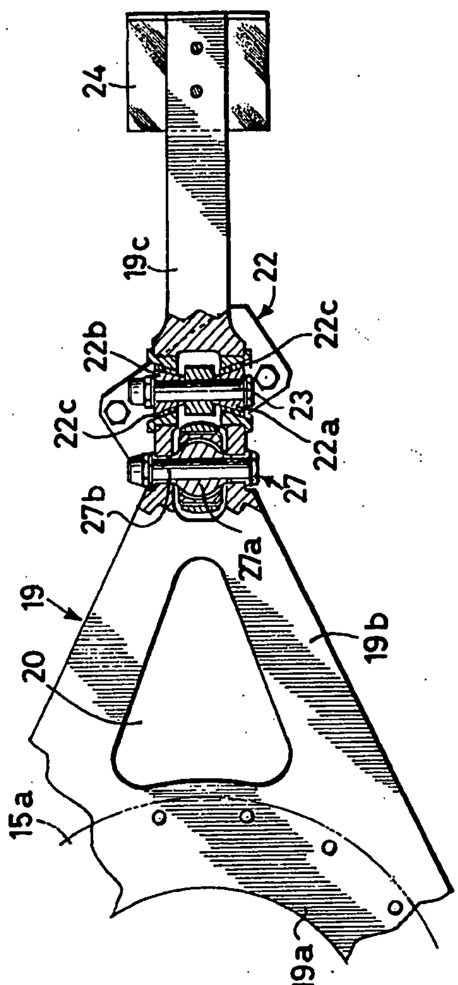
21
19c
22
23
27b
27
25
19b
15a
19a
15

FIG.5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19c
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22c
22a
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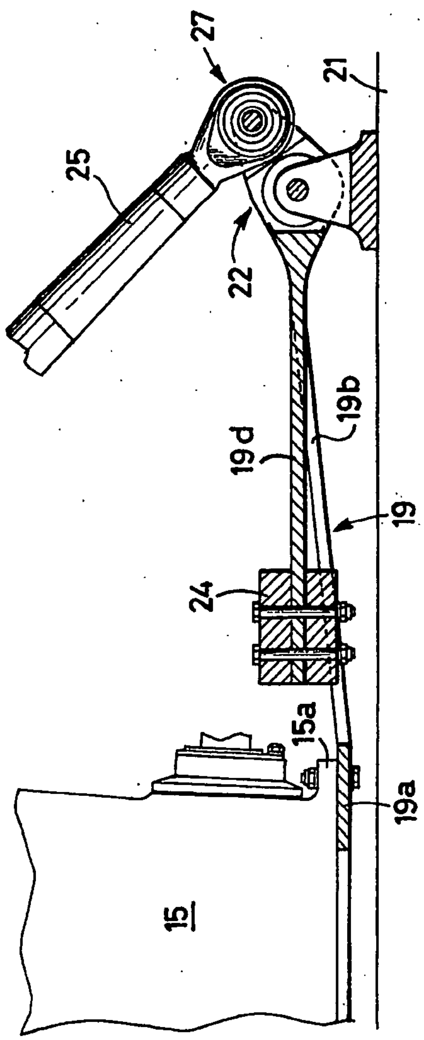
27b
27a
19
20
19b
15a
19a

FIG.6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25
21
22
19d
19b
19
24
15a
19a
15

FIG. 7

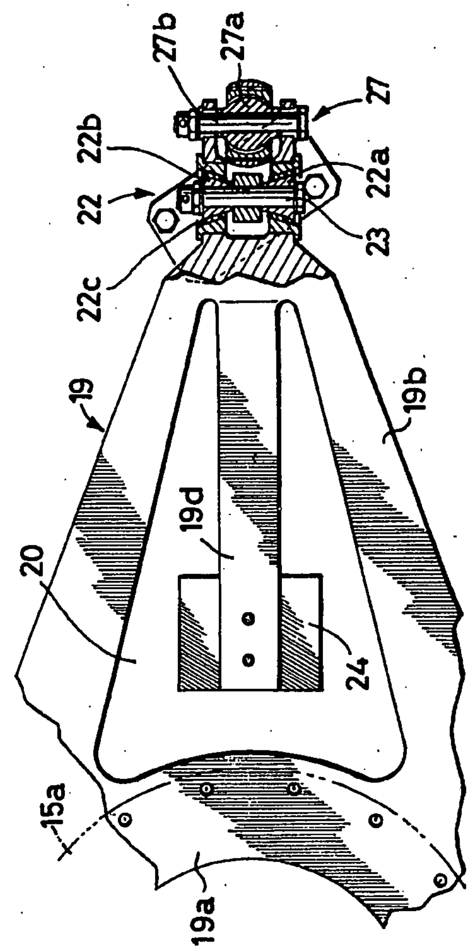
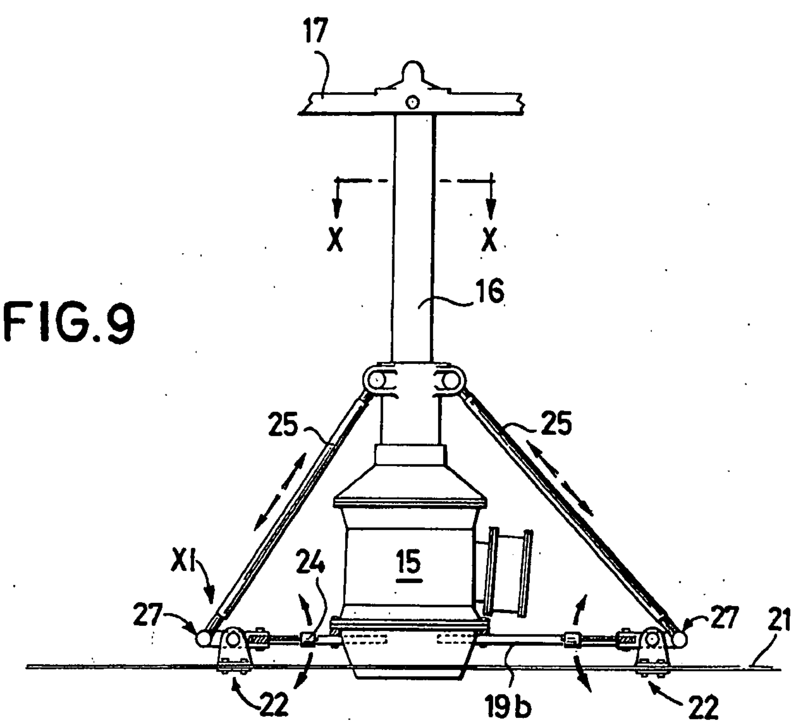

27b
27a
27
22b
22
22a
22c
23
19
19b
20
19d
24
15a
19a

FIG. 8

FIG.9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X
X
16
25
25
15
XI
24
27
27
21
22
19b
22

FIG.10

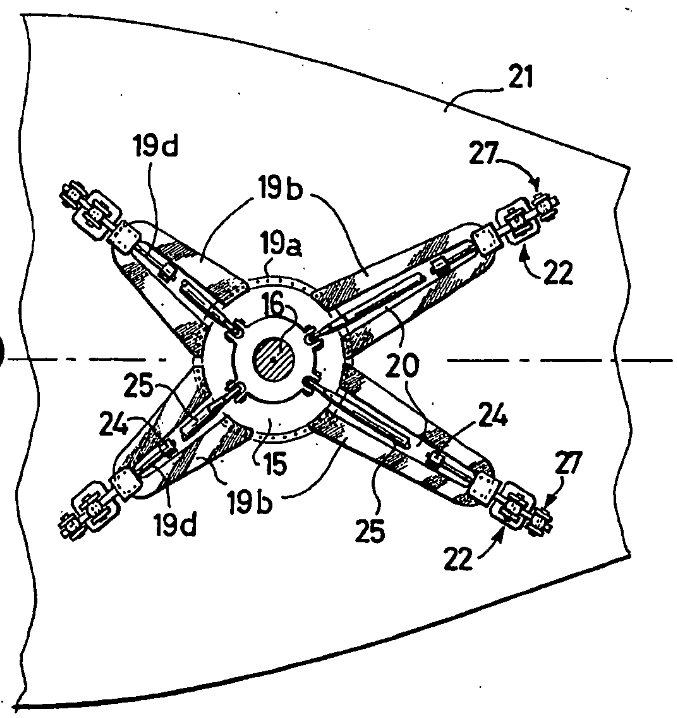
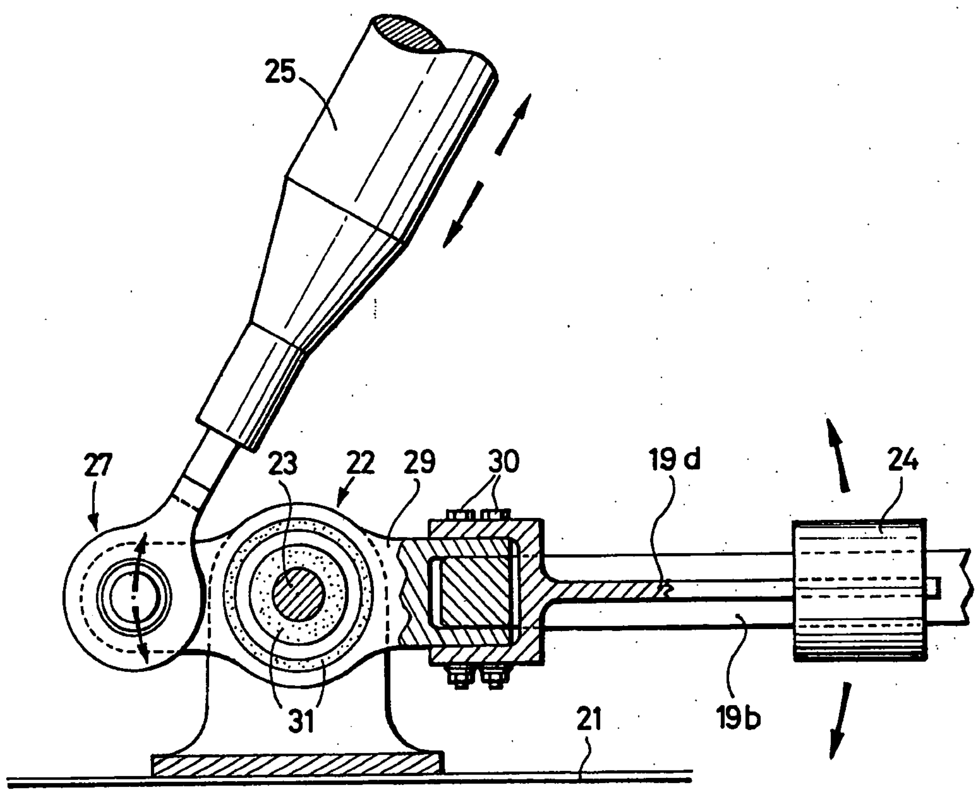

21
19d
27
19b
19a
22
16
20
25
24
24
27
15
19b
19d
25
22

FIG.11



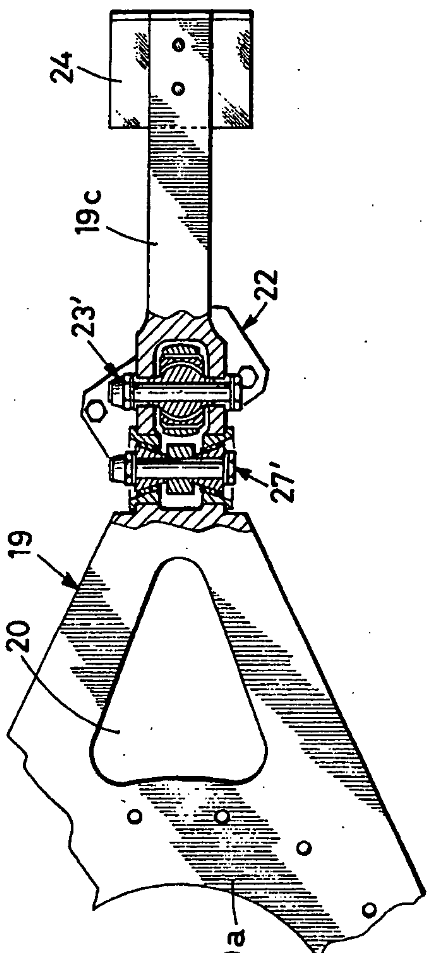



FIG.12

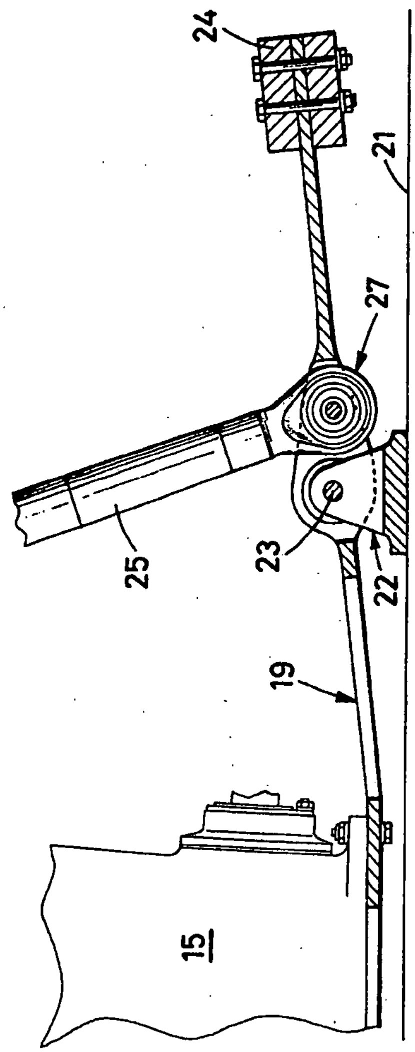



FIG.13

# ANTIRESONANT SUSPENSION DEVICE FOR HELICOPTER

This is a continuation of Ser. No. 229,749 filed Jan. 29, 1981 now abandoned.

## BACKGROUND OF THE INVENTION

In mechanical-drive helicopters, the lifting rotor is connected via a vertical transmission shaft to a speed reduction gear ensuring the transmission of power from the propulsion engine or engines of the apparatus. The assembly of the rotor-shaft-reduction gear constitutes, with respect to the structure of the helicopter, a relatively rigid assembly.

The speed reduction gear constitutes the main gear box and the case of each box is generally fixed on the structure of the helicopter by two types of link. The first link comprises a certain number of rigid elements, such as bars or stiffeners, in a number at least equal to three, interposed obliquely between the upper part of said case and strong points of the structure of the fuselage. The second link is a direct link between the lower part of this case and said structure; it is intended to take up the countertorque of the rotor, totally or partially. On the other hand, the functioning of said rotor for lifting and propelling the helicopter produces a vibration excitation essentially in the plane of said rotor. To attenuate its transmission to the fuselage, there may be disposed between the lower part of the case of the gear box and the fuselage, elements flexible in translation in a plane parallel to that of the rotor, but having considerable ridigity in rotation about its axis so as to be in a position to transmit the countertorque of the rotor.

U.S. Pat. No. 3,502,290 describes such an elastic link. It is formed by a flexible mounting plate constituted by a plate in which certain zones have cut-out portions, in the manner of a grid comprising bars. The flexibility thus obtained allows filtering of the vibrations by elastic deformation of the mounting plate at least in a direction thus privileged.

Other arrangements of suspension plate between the lower part of the main gear box and the top of the fuselage are described in U.S. Pat. No. 3,920,202 and in U.S. patent application Ser. No. 21,778 of Mar. 19, 1979 now U.S. Pat. No. 4,274,510. These mounting plates make it possible both to obtain, in their plane, a flexibility in mono- or multidirectional translation and to transmit the torque induced by the gear box on the fuselage.

All these devices ensure a relatively effective filtering of the vibrations produced by the rotor. However, on modern fast helicopters, the desired improvement in comfort imposes a search for a still lower vibration level. The corresponding attenuation of the vibrations may be obtained by means complementary of the preceding systems. For example, resonator devices employing blades or springs placed at different points of the helicopter fuselage and possibly on the flight controls may be used. These resonators oscillate at their own frequency, with such amplitude and phase that they absorb, at least partially, the alternate efforts transmitted by their points of fixation. These devices present, however, major drawbacks. On the one hand, such resonators must be disposed at each of the points where vibrations must be attenuated; this results in a considerable weight excess. On the other hand, the effect of attenuating the vibrations, obtained with each resonator, is limited to the vicinity of its point of fixation.



Finally, the positioning of a resonator at a determined point may have for effect to amplify the vibrations at other points of the apparatus.

To overcome these drawbacks, resonators have been made, disposed above the hub of the rotor, from which they receive their excitation directly. These resonators may be constituted by a flapping weight, returned into neutral position by antagonistic springs. Such resonators are for example the subject matter of U.S. application Ser. No. 9614, now U.S. Pat No. 4,255,084, and No. 9578, now U.S. Pat. No. 4,281,967 of Feb. 5, 1979. These resonators offer the advantage of opposing the vibrations as near as possible to their source, i.e. on the rotor hub itself. They are all the more efficient as they are used in complement to the devices for suspension of the lower part of the main gear box, such as those described in U.S. Pat. Nos. 3,502,290 and 3,920,202.

However, such antivibration devices (hub resonators and elastic links between gear box and fuselage), although they generally enable a very good vibration level to be obtained on helicopters having more than two blades on the main rotor, present the drawback of a relatively considerable weight and also of a high cost price.

## SUMMARY OF THE INVENTION

The present antiresonant suspension device remedies these drawbacks. It combines an elastic link between the lower part of the main gear box and the fuselage and flapping weights which are directly associated therewith, simply and in a compact assembly. This device is efficient in combatting the vibrations from the rotor not only for the longitudinal and transverse excitations, but also for the so-called vertical pumping excitations.

To this end, the present invention relates to an antiresonant suspension device for helicopter composed of a fuselage, a propulsion assembly, a rotor and a gear box located between said propulsion assembly and said rotor and aligned on the axis of the latter, this device comprising a set of at least three oblique connecting bars and a mounting plate for suspension of the gear box on the fuselage structure. According to the invention, the mounting plate comprises a central part fast with the bottom of the gear box and radial extensions in a number equal to the number of said oblique bars, forming coplanar arms disposed in star form around said central part, which are substantially rigid in their plane but flexible in a direction perpendicular thereto, each of these arms being connected, in the region of its end, on the one hand to the fuselage at a strong point thereof, on the other hand to the lower end of the corresponding connecting bar, by respective joints allowing it displacements by deformation in a direction perpendicular to its plane, and the end of each flexible arm being fast with a relatively rigid lever bearing, at its free end, a flapping weight.

In one embodiment, the joint by which each flexible arm is connected to the fuselage is a bearing joint whose axis is contained in the plane of the mounting plate and perpendicular to the longitudinal direction of said arm and to the axis of rotation of the rotor, whilst the joint by which each flexible arm is connected to the lower end of the corresponding connecting arm is preferably a ball joint. In another embodiment, the joint by which each flexible arm is connected to the fuselage is a ball joint, whilst the joint by which each flexible arm is connected to the lower end of the corresponding connecting arm is preferably a bearing whose axis is con-

tained in the plane of the mounting plate and perpendicular to the direction of said arm and to the axis or rotation of the rotor, said bearing advantageously being constituted by ball or roller bearings in order to reduce damping by friction.

The flexible arms advantageously have a width which decreases from the central part of the mounting plate towards their end, and preferably comprise an inner opening extending along their longitudinal axis.

In one embodiment of the device according to the invention, the lever bearing the flapping weight of each flexible arm is constituted by an extension of said arm directed outwardly, the flapping weight being located beyond the joint on the fuselage with respect to the centre of the mounting plate.

In another embodiment, the lever supporting the flapping weight mentioned above is constituted by an extension of the flexible arm directed inwardly, the flapping weight then being located in the opening in said arm, between the joint on the fuselage and the centre of the mounting plate.

The joint connecting each flexible arm to the fuselage is in principle located between the flapping weight and the joint connecting said arm to the corresponding connecting bar. The joint connecting each flexible arm to the corresponding connecting bar may also be provided to be located between the flapping weight and the joint connecting said arm to the fuselage.

The lever bearing each flapping weight may be integral with the corresponding flexible arm. It may also be connected thereto.

## BRIEF DESCRIPTION OF THE DRAWINGS

The invention will be more readily understood on reading the following description with reference to the accompanying drawings, in which:

FIGS. 1 and 2 schematically show, in elevation and in plan view respectively, a suspension device according to the invention in a first embodiment.

FIGS. 3 and 4 similarly show a device according to the invention in a second embodiment.

FIGS. 5 and 6 show in detail, on a larger scale, an embodiment of one of the flexible arms as shown respectively in FIGS. 1 and 2.

FIGS. 7 and 8 similarly show an embodiment of one of the the flexible arms as shown respectively in FIGS. 3 and 4.

FIGS. 9 and 10 shows, in elevation and in plan view, respectively, a practical embodiment of a device according to the invention.

FIG. 11 shows, on a larger scale, the detail XI of FIG. 9.

FIG. 12 is a view similar to FIG. 6 showing an interchange in the types of joints employed to secure the flexible arms of the suspension device; and

FIG. 13 is a view similar to that of FIG. 5 showing the interchange illustrated in FIG. 12.

## DESCRIPTION OF THE INVENTION

Referring now to the drawings, FIG. 1 shows the main gear box **15** of a helicopter, interposed between a propulsion assembly shown in dot-dash outline and the shaft **16** of the rotor of which only the hub **17** has been shown. Elements **15,16** and **17** are aligned on the axis or rotation **18** of the rotor.

The bottom **15***a* of the case of the gear box **15** is fixed by a ring of volts to the central part **19***a* of a flat, flexible mounting plate **19**, of which the plane is perpendicular to the axis **18** and which comprises radially extending, four in number in the present example, flexible arms **19***b* distributed regularly in star form around the central part **19***a*, with which they are integral (FIG. 2). The width of these arms **19***b* decreases from the central part **19***a* to their outer end and they have an inner longitudinal opening **20** of substantially triangular form.

The flexible arms **19***b* abut, in the region of their outer ends, on strong points of the upper part of the fuselage **21** of the helicopter, via bearings **22** of which the pivot pins **23** are contained in the plane of the mounting plate **19** and are perpendicular to the radial direction of the respective arms, as well as to the axis **18**. These bearings **22** are constituted so that they present virtually no damping nor friction and that, on the one hand, they may transmit to the fuselage the shearing stresses and the bending moments of which the arms **19***b* are the seat in their plane, whilst, on the other hand, they allow said arms slight angular displacements by their ends pivoting about pivot pins **23**, as shown in dashed lines in FIG. 1.

Each flexible arm **19***b* extends beyond the bearing **22** by a more rigid terminal part **19***c* forming a lever, which bears at its end a flapping weight **24**, the assembly **19***c*, **24** forming an antiresonant system acting by inertia.

The main gear box **15** is also connected to the structure of the fuselage **21**—indirectly—by means of a set of oblique bars **25**. These are substantially concurrent at their top ends at a focusing point **26** located at the top of the gear box **15**, on axis **18**, and they are connected at their lower ends, via a ball joint **27**, each to a corresponding flexible arm **19***b*, at a point thereon near the bearing **22** and nearer the axis **18** than said bearing.

In the present example, the oblique bars **25** are four in number, like the flexible arms **19***b*. More generally, it will be considered that the number of flexible arms of the mounting plate **19** is always equal to the number of oblique bars **25**, this number never being less than three.

The reaction of the drive torque of the rotor passes entirely from the bottom **15***a* of the gear box **15** into the fuselage **21** via the arms **19***b* of the flexible mounting plate **19**, which work in flexion in their plane. The horizontal shearing stresses produced in the plane of the rotor and transmitted by the shaft **16** and the gear box **15** are taken up on the structure of the fuselage by the flexible arms **19***b* working in their plane in traction or in longitudinal compression. These stresses do not substantially deform the arms **19***b*. On the contrary, the moments **28** of horizontal axis in the plane of the rotor and the moments due to the above-mentioned shearing stresses are also transmitted to the structure **21** by the arms **19***b*, but with attenuation, as they inflict thereon bending deformations in a direction parallel to the axis **18**, as indicated in dashed lines in FIG. 1.

The vertical lifting efforts of the rotor as well as the horizontal bearing reactions due to the efforts in the plane of the rotor and located at the level of the upper attachment of the oblique bars **25** load said bars longitudinally and are transmitted to the structure of the fuselage **21** via joints **27** and **22**.

Due to the geometry of the assembly, it is seen that the dissymetrical efforts in the plane of the rotor provoke on each arm **19***b*, between the lower joint **27** of the bar **25** and the adjacent bearing **22**, a moment which accentuates the deformation of the arm caused by the bending stress applied correlatively by the bottom **15***a* of the gear box **15**. Thus, the longitudinal and lateral dynamic efforts and moments (with respect to the direction of flight of the helicopter) produced in the plane of

the rotor with frequency b Ω (b being the number of blades of the rotor and Ω its speed of rotation), which the present device is essentially to attenuate, as well as the dynamic vertical "pumping" efforts, are filtered because the elastic reactions at the points of attachment **22** on fuselage corresponding to the deformations of the flexible arms of the mounting plate are counterbalanced by the forces of inertia developed by the flapping weights and which give said points of attachment reactions of direction opposite the direction of the above-mentioned reactions. This effect completes the filtering action of the flexible mounting plate **19**.

In the variant embodiment shown in FIGS. **3** and **4**, elements homologous to those of FIGS. **1** and **2** bear the same references. The only differences reside in that the outer extensions **19***c* of the arms **19***b* are replaced by inner extensions **19***d*, i.e. directed towards the central axis **18**, the weights **24** flapping through the openings **20** in the arms **19***b*, and that the ball joints **27** of the connecting bars **25** are located beyond the bearing joints **22**, i.e. further from axis **18** than said bearin joints. Such an assembly, which is more compact than that of FIGS. **1** and **2**, functions in the same manner.

FIGS. **5** and **6** show a practical embodiment of the device of FIGS. **1** and **2**, and FIGS. **7** and **8** a practical embodiment of the device of FIGS. **3** and **4**. In the devices thus produced, the central base plate **19***a* of the flexible mounting plate **19**, its flexible arms **19***b*, the extensions **19***c* or **19***d* thereof and the supports of the ball joints **27** and bearing joints **22**, are constituted by a monobloc piece of forged metal. The lower ends of the oblique bars **25** are respectively attached to the mounting plate **19** by a self-lubricating ball joint **27***a* through which a pin **27***b* passes. In the bearing joints **22**, the pin **23** abuts on a double conical bearing **22***a, b* with interposition of an elastomer layer **22***c*, so as to allow the slight angular displacements desired, with negligible friction.

FIGS. **9**, **10** and **11** show a practical embodiment of the device of FIGS. **3** and **4**. Here, the flexible arms **19***b* are composed of metal blades bolted on a central base **19***a* fixed to the bottom of the case of the main gear box **15**. The longitudinal openings **20** in these blades make it possible to house the flapping weights **24** mounted at the ends of flexible blades **19***d* added on the arms **19***b*, via a piece **29** forming the end of each of the arms **19***b*, by means of bolts **30** passing through the whole. Each piece **29** comprises the bearing joint **22** for connection to the fuselage **21**, here constituted by a laminated sleeve joint comprising two elastomer layers **31** around axis **23**, and the ball joint **27** for connection to the corresponding oblique bar. In this embodiment, the flexible arms **19***b* are made of metal and, for example, cut out from a thick metal sheet. In a preferred embodiment, they may also be made of a laminated material made of high-resistance fibres embedded in a thermohardened synthetic resin.

FIGS. **12** and **13** show the interchangeability of the types and position of the joints **22** and **27**. As seen in FIG. **12**, the joint **27'** between the oblique bar **25** and the arm **19** is a double conical bearing such as described earlier for joint **22**, while the joint **23'** between the arm **19** and the fuselage is a ball joint. In FIG. **13** the joint **27** between the oblique bar and the arm **19** is on the exterior of the joint **22**.

What is claimed is:



**1**. An anti-resonant suspension system for supporting the rotor and gear box of a helicopter on the fuselage thereof comprising a mounting plate having a central part on which is secured the bottom of the gear box and a plurality of radially extending flexible arms disposed coplanar uniformly about the central axis of said gear box, said arms being rigid within their plane and flexible in a direction perpendicular thereto, each of said arms being joined at its end to said fuselage, a plurality of oblique connecting bars corresponding in number to the number of said arms, said bars being joined at one end to the upper end of said gear box and at its other end to a respective one of said arms at a point along said arm adjacent to but spaced latterally from the joint between said arms and said fuselage, said respective joints permitting flexing of each of said arms along their length by deformation in a direction perpendicular to the plane of said arms, each of said arms having a lever integral with its outer end and extending freely therefrom in the plane of said arm, said lever having a flapping weight secured to the free end thereof substantially coplanar with said arm.

**2**. The device of claim **1**, wherein the joint by which each flexible arm is connected to the fuselage is a bearing joint whose axis is contained in the plane of the mounting plate and perpendicular to the longitudinal direction of said arm.

**3**. The device of claim **2**, wherein the joint by which each flexible arm is connected to the lower end of the corresponding connecting bar is a ball joint.

**4**. The device of claim **1**, wherein the joint by which each flexible arm is connected to the fuselage is a ball joint.

**5**. The device of claim **4**, wherein the joint by which each flexible arm is connected to the lower end of the corresponding connecting bar is a bearing joint.

**6**. The device of claim **1**, wherein the flexible arms have a width which decreases from the central part of the mounting plate towards their end.

**7**. The device of claim **1**, wherein the flexible arms comprise an inner opening extending along their longitudinal axis.

**8**. The device of claim **7**, wherein the lever bearing the flapping weight of each flexible arm is constituted by an extension of said arm directed towards the inside, the flapping mass being located in the opening of the flexible arm, between the joint on the fuselage and the centre of the plate.

**9**. The device of claim **1**, wherein the lever bearing the flapping weight of each flexible arm is constituted by an extension of said arm directed towards the outside, the flapping weight being located beyond the joint on the fuselage with respect to the centre of the plate.

**10**. The device of claim **1**, wherein the joint connecting each flexible arm to the fuselage is located between the flapping weight and the joint connecting said arm to the corresponding connecting bar.

**11**. The device of claim **1**, wherein the joint connecting each flexible arm to the corresponding connecting bar is located between the flapping weight and the joint connecting said arm to the fuselage.

**12**. The device of claim **1**, wherein the lever bearing each flapping weight is connected to the corresponding flexible arm.

* * * * *

March 26, 1935. R. A. MONROE 1,995,620

VIBRATION INHIBITOR

Filed July 9, 1932

Fig. 1.

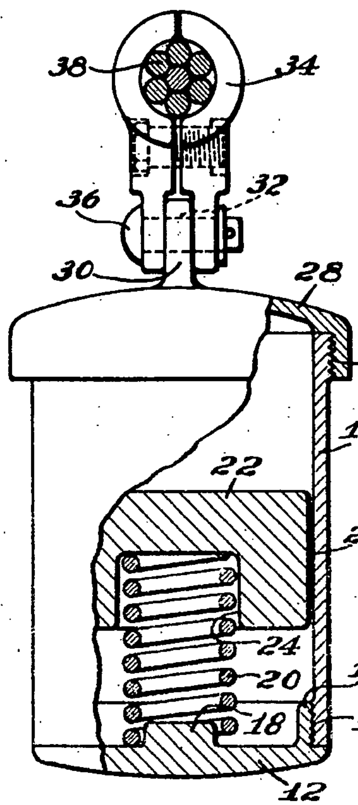



Fig. 2.

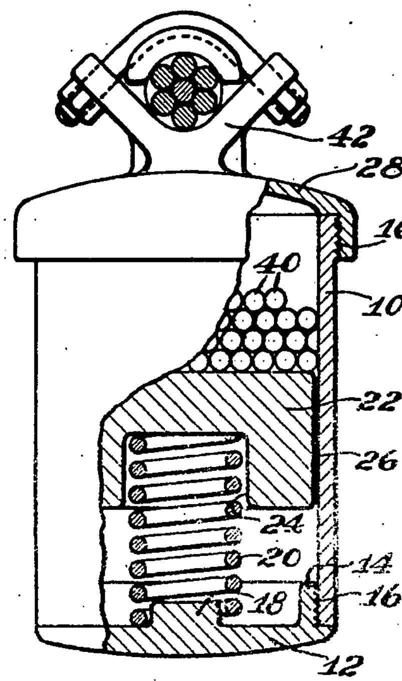
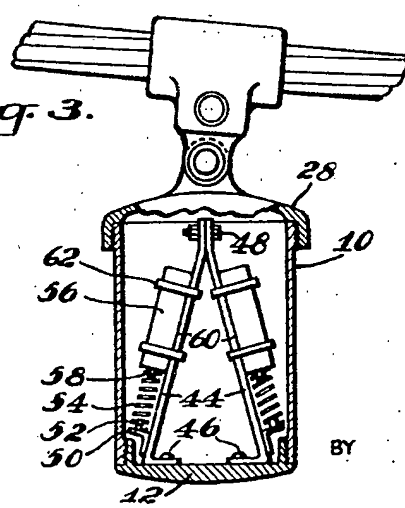



Fig. 3.




INVENTOR

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ATTORNEY

# UNITED STATES PATENT OFFICE

1,995,620

## VIBRATION INHIBITOR

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Application July 9, 1932, Serial No. 621,565

8 Claims. (Cl. 173—13)

The invention relates to vibration inhibitors primarily adapted for use in connection with overhead transmission lines.

While the phenomenon of vibration of overhead conductors is not yet thoroughly understood, the damaging results of vibration are quite generally known and considerable effort has been expended in the direction of inhibiting vibration. Vibration has caused damage to conductors in widely scattered locations throughout the country, and although a very small percentage of the total mileage of transmission lines has been adversely affected, the threat of a shortening of the life of the conductor by vibration has directed attention to the solution of the problem.

Generally speaking, all transmission lines, regardless of material, size of conductor, span length, tension or character of supports, vibrate under certain conditions. The vibration may vary all the way from a slow vertical oscillation of the span, as one loop with an amplitude of several feet which is generally termed "dancing", to the audible high frequency singing of small wire such as is often heard on telephone lines. The dancing or whipping of overhead conductors is quite distinct from the phenomenon of resonant vibration, toward the elimination of which the present invention is primarily directed.

It is well known that a suspended cable can be caused to vibrate in a vertical plane in a series of nodes and loops by the action of a wind blowing transversely to the line. The generally accepted theory explaining this is that air currents moving across the line create periodic eddies on the leeward side of the conductor. These eddies create rapidly alternating condensations and rarefactions which exert minute vertical forces on the conductor. When the period of these eddies coincides with the natural period, or a harmonic, of the elastic conductor, a resonant condition exists and the motion builds up until balanced by internal and external damping influences. The maximum amplitude of vibration rarely exceeds the diameter of the conductor and ordinarily is less. The frequency may vary from a minimum of a few cycles per second for large cables, to an upper limit for small wires which lies within the range of audibility.

Sharp fatigue fractures of wire at insulator ties and other points of support have occurred with all conductor material ever since the beginning of overhead line construction. These failures are generally recognized to be the result of fatigue occasioned by vibration. In the installation of high tension transmission lines, with the growing tendency to increase the length of span and also to increase the tension in the cable, increasing trouble resulting from resonant vibration has been encountered. Long continued vibration may weaken the metal of the cable by fatigue and ultimately lead to rupture at its point of support. The number of cycles or repetitions of stress necessary to reach the fatigue limit of the metal of the cable varies considerably with the type of cable employed. For example, only 10 to 20 million cycles are necessary to reach the fatigue limit for ferrous metals, while in the case of a non-ferrous metal such as aluminum, from 400 to 500 million cycles may be necessary. Experience indicates that the completion of 10 to 20 million cycles of vibration will occur comparatively early in the life of a cable. It is quite manifest that under these conditions the problem of eliminating vibration assumes serious proportions, and it is the solution of this problem to which the present invention is directed.

It is a general object of my invention to provide an improved form of vibration inhibitor which virtually eliminates the vertical vibrations that occur in transmission lines and consequently obviates many of the failures that are attendant upon such vibration. Toward this end the invention broadly contemplates the provision of a resiliently suspended mass or inertia member with means for attaching the same to a cable at some predetermined point intermediate the ends of the span. This resiliently-suspended inertia member is of sufficient weight to offer a considerable amount of resistance to suddenly-applied forces resulting from the tendency of the cable to vibrate in a vertical plane. Associated with the inertia member are frictional means so arranged that extremely minute vibrations represented by relative movement between the cable and the inertia member will develop friction with the consequent absorption of energy from the vibrating line. The inertia member, frictional means, and means of resiliently suspending the same from the cable are contained and embodied in a compact unitary structure protected by its design from adverse weather conditions.

In designing the improved vibration inhibitor consideration has been given to preventing the formation of corona discharge at the point of attachment of the inhibitor. Such consideration has resulted in a design of inhibitor by which the electrostatic flux in the medium surrounding the conductor is not materially disturbed.

The provision of a vibration inhibitor which is f simple, compact and unitary construction

and which consequently is unlikely to get out of order, yet one which may be disassembled for the purpose of renewal of its parts or for repair, and one which is extremely efficient in its operation are further desiderata that have been borne in mind in the conception and reduction to practice of the present invention. Further objects and advantages of the invention will become more apparent as the nature of the invention is better understood, and it consists in a novel construction and arrangement of parts shown in the accompanying drawing in which:

Fig. 1 is a side elevation, partly in central vertical section, of an assembled vibration inhibitor constructed in accordance with the principles of the present invention, showing the same secured to a transmission line by suitable clamping means;

Fig. 2 is a view similar to Fig. 1, illustrating a modified form of the invention;

Fig. 3 is a side elevation, partly broken away in central vertical section, showing a further modification.

In the drawing like characters of reference are applied to designate like parts throughout. Referring first to Fig. 1, the vibration inhibitor therein disclosed involves in its general organization a cylindrical casing 10 having a bottom plate 12 provided with a flange 14 adapted for attachment to the lower portion of the casing in any suitable manner, as for example by means of the threaded connection 16. The plate 12 is provided with an internal centrally-located boss 18 which serves to center a coil spring 20 into the lower end of which the boss 18 projects. The coil spring 20 supports an inertia member 22 having a recess 24 into which the upper end of the coil spring 20 extends. The inertia member is preferably formed of a relatively heavy metal or alloy, as for example iron or brass. The diameter of the inertia member is very slightly less than the internal diameter of the casing 10 so that an annular clearance space 26 exists between the inner walls of the casing and the circumferential surface of the inertia member 22. I have found that for a typical installation a clearance 26 on the order of 0.005 inch is satisfactory.

The upper end of the casing 10 is closed by means of a plate 28 secured to the casing 10, as by means of the screw thread connection 16. The closure plate 28 as well as the bottom closure plate 12 is slightly rounded in configuration. This feature, together with the fact that the casing 10 is substantially cylindrical, prevents formation of serious corona discharge as might otherwise be occasioned. The upper closure plate 28 is provided with an upstanding lug 30 having a transverse aperture 32 extending therethrough. A 2-piece separable clamp, designated generally by the reference numeral 34, is attached to the lug 30 as by means of a pin 36, providing suitable means for suspending the vibration inhibitor from the transmission line 38.

In use the device is secured to the cable at some predetermined point intermediate the ends of the span. Usually it is preferable to employ two of the vibration inhibitors, one of which is positioned near each of the insulators or points of suspension. For unusually long spans more than two inhibitors may be required.

The operation of the improved inhibitor disclosed in Fig. 1 is as follows: The casing 10 being rigidly secured to the cable 38 must necessarily move in a vertical direction in unison with the cable at the point of attachment of the casing. Such vertical movement of the casing is effectively damped against the inertia member 22 which, being of considerable mass, resists initial movement vertically. This damping action is effected partially by reason of th clearance between the inertia member and the casing 10 which, as the casing moves up or down relative to the inertia member, allows the air contained within the casing to be forced from one side of the inertia member to the other side through the clearance space 26. The frictional resistance of the air in passing through the space between the walls of the casing and the inertia member results in an absorption of energy which exerts an effective damping action. Although the hysteresis of the spring 20 is extremely small, nevertheless the damping action is further, though slightly, increased by virtue of the energy absorbed by the spring which resists any tendency of the cylinder to move upwardly with respect to the inertia member 22.

In Fig. 2 there is shown a modified form of vibration inhibitor. The details of construction of the casing 10, plate 12, inertia member 22 and compression spring 20 remain substantially the same as in the form of the invention just described in connection with Fig. 1. The presence of lead or steel shot designated at 40, which is contained within the casing 10 in the space above the inertia member 22, is the distinguishing characteristic of this form of the invention. As an equivalent of the shot specifically referred to, sand, gravel and viscous liquid or loose particles contained in liquid, or other loose material that will offer frictional resistance to the vibration of the casing may be employed. Thus, when the transmission line 38 is vibrated vertically the casing, vibrating in unison therewith, and therefore vibrating relative to the inertia member will cause the particles 40 to be rubbed together and against the walls of the casing, producing friction between the particles themselves and between the particles and the walls of the casing, thus serving to absorb the energy of vibration. In actual use I have found that the inhibitor prevents the occurrence of vibration of any appreciable magnitude. The benefits conferred by my invention result from the inhibiting or damping action of the device herein disclosed. The maximum amplitude of vibration is thereby reduced to a point which virtually eliminates fatigue stresses.

A modified form of clamping device 42 is shown in connection with this form of vibration inhibitor. This clamp 42 is characterized by smooth lines which materially aid in preventing the formation of undesirable corona discharge.

Referring now to Fig. 3, wherein a further embodiment of the invention is shown, the casing 10, bottom closure plate 12 and top closure plate 28 are substantially the same as these respective elements in Fig. 1. A pair of inclined plates 44 have their lower ends secured to the closure member 12 as at 46. The upper edges of the plates are secured together at 48 by any suitable means. Each plate 44 is provided with a transversely-extending angle piece 50 adjacent its bottom having a centering projection 52 about which is disposed the lower end of a coil spring 54. A movable inertia member 56 having a centering projection 58 at its lower end is supported on each spring 54. This inertia member is provided with a flat face 60 for frictional engagement with the surface of the respective inclined plate

44. As the casing vibrates in a vertical direction the inertia members 56 tend to remain in their initial position and therefore move with respect to the inclined plates 44, producing friction between the contacting surfaces to absorb the energy of vibration. Suitable means may be provided for guiding the inertia members 56 with respect t the inclined plates 44 and for this purpose I have shown the guiding straps 62.

In the interest of clarity I have described my invention with reference to particular embodiments and have employed specific language. In the use of such language I have no intention of excluding any equivalents or minor variations of the invention set forth.

What I claim is:

1. In a vibration inhibitor for transmission lines, a casing member adapted to be attached to a cable span, said casing member having a detachable base member, an inertia member within said casing, said inertia member being positioned on and removable with said detachable base member.

2. In a vibration inhibitor for transmission lines, a cylindrical casing member adapted to be attached to a cable span, said casing member having a detachable base member, an inertia member slidably mounted within said casing, said inertia member being resiliently supported on said detachable base and removable therewith.

3. In a vibration inhibitor for transmission lines, a cylindrical casing member adapted to be attached to a cable span, said casing member having a detachable base member, an inertia member slidably received within said casing and provided with a centrally located recess, a coil spring having one of its ends positioned within said recess, a boss extending upwardly from said base member, said boss being positioned within the other end of said coil spring, said coil spring and inertia member being removable as a unit on and with the removal of the detachable base member.

4. In a vibration inhibitor for transmission lines, a cylindrical casing member having removable top and base members, said top and base members having rounded configurations so as to eliminate corona discharge, means associated with the top member for attaching the casing to a cable span, an inertia member within said cylindrical casing, said inertia member being slightly smaller in diameter than the interior diameter of the casing, an upwardly projecting boss on the base member, a recess in said inertia member, and a coil spring centered on said boss and extending into said recess, whereby the inertia member is resiliently supported and positioned on said base member and removable therewith on detachment of said base.

5. In a vibration inhibitor for transmission lines, a cylindrical casing member adapted to be attached t a cable span, said casing member having a detachable base member, an inertia member slidably received within said casing and provided with a centrally located recess, a coil spring having one of its ends positioned within said recess, a boss extending upwardly from said base member, said boss being positioned within the other end of said coil spring, said coil spring and inertia member beng removable as a unit on and with the removal of the detachable base member, and friction-producing means in the form of a mass of relatively small particles disposed in said casing and supported upon said inertia member.

6. In a vibration inhibitor for transmission lines, a cylindrical casing member adapted to be attached to a cable span, said casing member having a detachable base member, an inertia member supporting friction-producing means slidably received within said casing and provided with a centrally located recess, a coil spring having one of its ends positioned within said recess, a boss extending upwardly from said base member, said boss being positioned within the other end of said coil spring, said coil spring and inertia member being removable as a unit on and with the removal of the detachable base member.

7. In a vibration inhibitor for transmission lines, a cylindrical casing adapted to be attached to a cable span, said casing having a detachable base member, an inertia member slidably mounted within said casing and spaced from the wall of said casing to provide a frictional passage for air contained within said casing, and a yieldable element disposed between said detachable base and inertia member, said inertia member, yieldable element, and base being removable from the casing as a unit.

8. In a vibration inhibitor for transmission lines, a cylindrical casing member adapted to be attached to a cable span, said casing member having a detachable base member, a member attached to said base interiorly of said casing and providing an inclined surface, an inertia member mounted in frictional contact with said inclined surface, and a yieldable means for supporting said inertia member on said inclined surface.

ROBERT A. MONROE.

# United States Patent [19]

**Shtarkman**

[11] **Patent Number: 4,509,730**

[45] **Date of Patent: Apr. 9, 1985**

[54] **FLEXIBLE WALL SPRING DAMPER**


[75] Inventor: **Emile M. Shtarkman**, Euclid, Ohio

[73] Assignee: **Imperial Clevite Inc.**, Glenview, Ill.



[21] Appl. No.: **436,330**

[22] Filed: **Oct. 25, 1982**

[51] **Int. Cl.**³ .............................................. **F16F 13/00**

[52] **U.S. Cl.** ...................................... **267/35**; 188/268; 267/64.25; 267/64.27; 267/140.1; 267/140.3; 267/141; 267/152; 267/153; 267/122; 280/662; 280/697; 280/712; 280/716

[58] **Field of Search** .................. 188/268; 267/35, 152, 267/140.3, 141, 140.1, 153, 64.25, 64.27, 122; 280/697, 662, 712, 716


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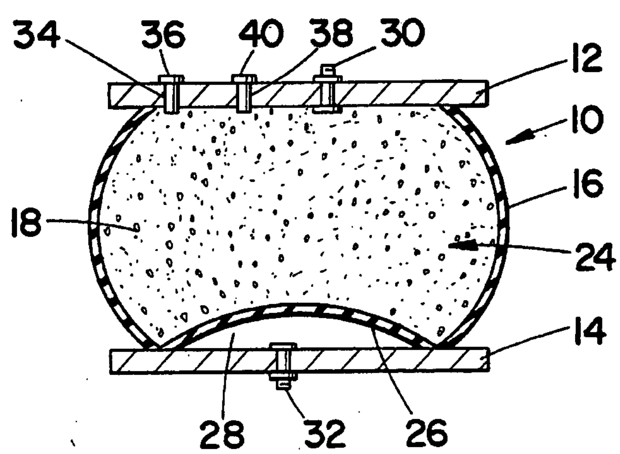
[57] **ABSTRACT**


A flexible wall spring damper including opposed first and second rigid mounting members interconnected by a flexible side wall is provided. A spring damper chamber is included within the flexible wall and is interposed between the opposed rigid mounting members. A plurality of elastomeric particles are included in the damper chamber. A selectively pressurizable gas chamber is interposed between the rigid mounting members and the flexible side walls. In one embodiment of the invention, a flexible diaphragm connected to one of the rigid mounting members defines the selectively pressurizable gas chamber. In another embodiment of the invention, a spiral tube defines the flexible side wall and includes an inner chamber including the plurality of elastomeric particles. Relative movement between the opposed rigid mounting members operates to stress the flexible side wall and the plurality of elastomeric particles, and vary the volume of the selectively pressurizable gas chamber. The invention exhibits the operational characteristic of a constant natural frequency independent of the mass supported by the device.


**15 Claims, 3 Drawing Figures**



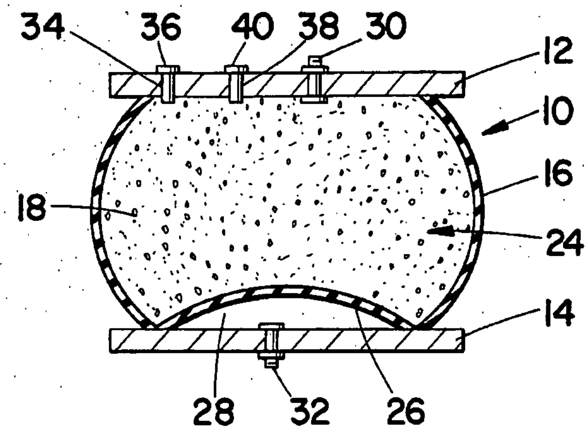



FIG. 1

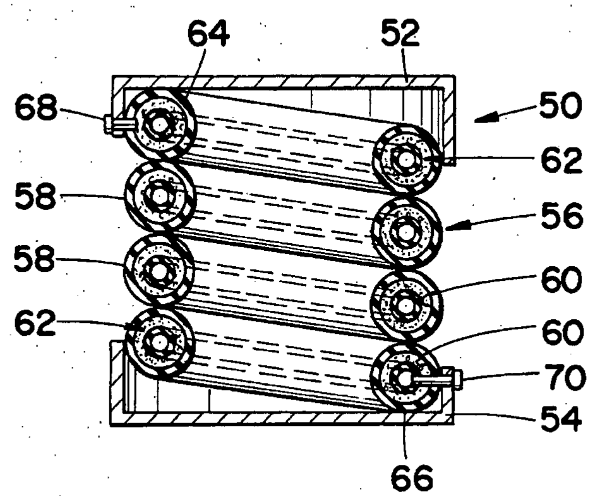



FIG. 2

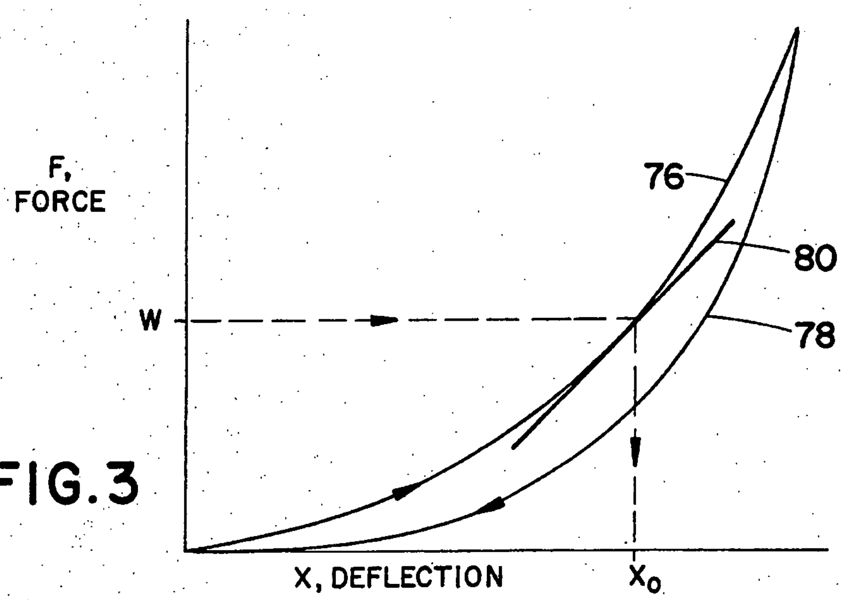



FIG. 3

# FLEXIBLE WALL SPRING DAMPER

## BACKGROUND OF THE INVENTION

The present invention relates generally to load carrying shock absorbers. More particularly, it relates to shock absorbers of the type which are mounted to vehicles and which use both an elastomeric spring and a selectively pressurizable gas chamber for carrying and leveling load, absorbing shock, and energy dissipation. However, it will be appreciated by those skilled in the art that the invention can be readily adapted for use in other environments as, for example, where similar spring dampening devices are employed to protect or cushion other items.

Known spring damper devices of the type described have included elastomeric shear springs, elastomeric diaphragms, selectively pressurizable gas chambers and associated communicating fluid chambers including restrictive orifice means therebetween for restricting the flow of fluid between the fluid chambers. Such a spring dampening device is described in Application Ser. No. 208,013, filed Nov. 18, 1980, now abandoned, in the name of Shtarkman et al and assigned to the assignee of the subject application. In the Shtarkman et al application an expandable and contractable elastomeric bladder for separating a gas chamber from a fluid chamber in a viscous spring damper is provided. Varying the pressurization of the gas chamber is employed for varying the spring rate of the viscous spring damper, calibrating the damper or leveling a load supported thereby. In addition, fluid flow between the communicating fluid chambers absorbs and dampens the shock and spring forces occurring during operation of the device. Such a structure provides the advantageous operating characteristics of both a spring and a shock absorber in one package.

A particular problem inherent in viscous spring dampers including fluid chambers is the limitation of a short stroke. In other words, the extent of compressive reduction of the spring damper device may be unduly limited because the volume of fluid that must be displaced between communicating fluid chambers may become too great to allow compressive reduction of the device to a desired level. There may simply not be enough room for the fluid, which is typically an incompressible hydraulic fluid, to be displaced. As a result, failure of the viscous spring damper would occur upon compression of the device beyond a certain level, either occurring through fracture of the device's housing or shear spring, or through shear spring bond failure.

Another problem with prior viscous spring dampers is instability of the spring characteristics upon excessive deflection of the device. Specifically, an elastomeric shear spring is stressed primarily in shear upon deflection of a viscous spring damper. However, in situations where the viscous spring damper is stroked or compressed beyond a point where the shear spring becomes unstable, the shear spring operates irregularly. On a load/deflection curve, this is manifested as a drop in load with increasing deflection, rather than a steady increase in load with increasing deflection. Once in this position, it is difficult for the spring to expand or "flip back" rapidly. Such difficulty further results in slow response in the operation of the viscous spring damper.

Yet another problem with prior viscous spring dampers is that the natural frequency of spring response is dependent upon the mass supported by the device.



When employed in a vehicle suspension system, a non-constant natural frequency device translates into rider discomfort as a vehicle becomes loaded.

The present invention contemplates a new and improved dry viscous spring damper which overcomes all of the above referred to problems and others to provide a new viscous spring damper which is simple in design, economical to manufacture, readily adaptable to a plurality of load carrying and shock absorbing uses, and which provides improved shock absorption and energy dissipation.

## BRIEF SUMMARY OF THE INVENTION

In accordance with the present invention, there is provided a flexible wall spring damper including opposed first and second rigid mounting members interconnected by a flexible side wall for defining a flexible wall spring damper chamber. A plurality of elastomeric particles are included in the damper chamber. Relative movement between the opposed rigid mounting members operates to stress the flexible side wall and the plurality of elastomeric particles.

In accordance with another aspect of the present invention, there is provided a flexible wall spring damper including opposed first and second rigid mounting members interconnected by a flexible side wall for defining a flexible wall spring damper chamber. A plurality of elastomeric particles are included in the damper chamber. A flexible diaphragm is connected to the second rigid mounting member and comprises a selectively pressurizable gas chamber. Relative movement between the opposed rigid mounting members operates to stress the flexible side wall, the plurality of elastomeric particles and vary the volume of the gas chamber.

In accordance with another aspect of the present invention, the plurality of elastomeric particles have particle sizes of 30 mesh or smaller and further have a preferred aspect ratio of essentially 1 and highly irregular surfaces.

In accordance with a further aspect of the invention, the spring damper chamber further includes agglomeration inhibitors such as polytetrafluoroethylene particles, silica powders and silicone oil.

In accordance with the present invention, there is provided a spiral dry viscous spring damper including opposed rigid mounting members and an elastomeric spirally configured connecting member therebetween. The connecting member comprises a first spirally configured tube including a first plurality of tube windings and including a tube inner chamber. A plurality of elastomeric particles are included in the inner chamber. Relative movement between the rigid mounting members operates to stress the sprirally configured tube and the plurality of elastomeric particles.

In accordance with another aspect of the present invention, a selectively pressurizable elastomeric member is interposed between the mounting members and within the plurality of tube windings. At least one of the mounting members includes valve means for pressurizing the elastomeric member. The elastomeric member preferably comprises a second spirally configured tube including a second plurality of tube windings disposed within the first spirally configured tube.

One benefit obtained by use of the present invention is a flexible wall spring damper having improved operating characteristics.

Another benefit obtained from the present invention is an improved flexible wall spring damper having a plurality of elastomeric particles received in a spring damper chamber for load carrying and leveling, shock absorption, dampening and energy dissipation.

Another benefit obtained from the present invention is a flexible wall spring damper which provides significantly longer stroke of operation with improved load versus deflection stability and improved spring response.

Yet another benefit obtained from the present invention is a flexible wall spring damper which exhibits a constant natural frequency generally independent of the mass supported by the device.

Other benefits and advantages for the subject new flexible wall spring damper will become apparent to those skilled in the art upon a reading and understanding of the specification.

## BRIEF DESCRIPTION OF THE DRAWINGS

The invention may take physical form in certain parts and arrangement of parts, the preferred and an alternative embodiment of which will be described in detail in the specification and illustrated in the accompanying drawings which form a part hereof and wherein:

FIG. 1 is a cross-sectional elevational view of a preferred embodiment of a flexible wall spring damper constructed in accordance with the present invention;

FIG. 2 is a cross-sectional elevational view of an alternative embodiment of a flexible wall spring damper constructed in accordance with the present invention comprising a spiral dry viscous spring damper; and,

FIG. 3 is a graph showing the deflection characteristics of the present invention.

## DETAILED DESCRIPTION OF THE INVENTION

Referring now to the drawings wherein the showings are for purposes of illustrating the preferred and an alternative embodiment of the invention only and not for purposes of limiting same, FIG. 1 shows a flexible wall spring damper **10** including a first rigid mounting member **12** and an opposed second rigid mounting member **14**. The rigid mounting members are interconnected by a flexible side wall **16** which includes a flexible wall spring damper chamber **18**. Included in the damper chamber **18** is a plurality of elastomeric particles **24** and a flexible diaphragm **26**. Diaphragm **26** is connected to the second rigid mounting member **14** and may be an integral part of flexible side wall **16**. A selectively pressurizable gas or air chamber **28** is interposed between the second rigid mounting member **14** and the diaphragm **26**. Conventional valve means **30,32** are included in first and second mounting members **12,14** respectively to provide the supply of air or gas to spring damper chamber **18** and gas chamber **28**. A sealable filler port **34** is included in first mounting member **12** for the ingress and egress of elastomeric particles **24** to the spring damper chamber **18**. A port plug **36** (or valve) is employed to seal the port **34**. A sealable gas filter **38** is also included in first mounting member **12** for the filtered egress of gas or air from spring damper chamber **18**. Gas filter **38** may be conventionally sealed with a filter plug **40** (or a valve).

The flexible wall spring damper **10** has an overall generally cylindrical configuration and is preferably employed in a vehicle suspension system to carry and level load, absorb shock and dissipate energy.

It is a particular feature of the invention that spring damper chamber **18** includes a plurality of solid elastomeric particles **24** to be particularly useful as the working medium for the invention which are essentially incompressible in volume. Elastomeric particles **24** may be constructed of various elastomeric substances, preferably a natural rubber compound with a Shore A hardness range of 45 to 70, with an elongation-at-break of at least 500%, and with a carbon black loading such that the hysteretic properties are in the range of those normally used in automotive bushing compounds; and preferably comprise particles having a particle size of 30 mesh or smaller. The particles must be so sized that they act as a system of individual particles. At a particle size of 30 mesh or smaller, this occurs. Such a size range is best suited for employment in a spring damper device of the type of the invention. It is also preferred that the particles have an aspect ratio of essentially 1 and have highly irregular surfaces.

Maintaining the individuality of the particles **24** is important to the proper functioning of the invention. If the particles become agglomerated, it has been found that performance suffers. Therefore, additives have been utilized to inhibit agglomeration. Such additives tend to coat the particles and not remain free in the main chamber **18**. Particulate additives such as polytetrafluoroethylene powder and various types of silica which coat the elastomeric particles and have a mean particulate size much smaller than the elastomeric particles have been found to be particularly useful. Alternatively, a liquid additive, such as silicone oil, may also be employed. In addition, a thin film deposit from a gas could also be utilized.

Because the medium of chamber **18** is a plurality of elastomeric particulates **24**, packing efficiency is such that there are spaces between the individual particles. For non-compacted elastomeric particles with highly irregular surfaces, the particles may occupy less than 25% of the fill volume of chamber **18**. The balance of the volume of chamber **18** may be a vacuum, or be filled with air or other gases such as nitrogen.

The medium of elastomeric particles provides both a spring rate and a dampening capacity for the invention. The complex interactions of the particles which cause deformation of individual particles and movement of the particles relative to each other are what produces these behavior chacteristics. The spring and dampening rates can be tailored through appropriate choices of elastomeric particle types, sizes and shapes.

Spring damper chamber **18** may be completely filled with elastomeric particles as shown or could be only partially filled. However, when only partially filled, there must be sufficient elastomeric particles present in aggregation to fill the spring damper chamber upon relative movement between the mounting members **12,14** to an extent to dispose the members **12,14** in an equilibrium position upon being subjected to a static load.

With particular reference to FIG. 2, an alternative embodiment of the invention is shown. The alternative embodiment of a flexible wall spring damper comprises a spiral dry viscous spring damper **50** including a first rigid mounting member **52** and an opposed second rigid mounting member **54**. A flexible wall connecting member **56** is interposed between mounting members **52,54** and preferably comprises a first spirally configured tube including a first plurality of tube windings **58** and an inner chamber **60** extending the length of the spirally

configured tube. A plurality of elastomeric particles **62** are included in inner chamber **60**. Particles **62** are identical to the elastomeric particles employed in the embodiment of the invention of FIG. **1**. A first end portion **64** of the spiral tube, contiguous to the first rigid mounting member **52**, is bonded to the member **52** and a second end portion **66** of the spiral tube, contiguous to the second rigid mounting member, is bonded to the second member **54**. Conventional port means **68,70** are provided for supplying the elastomeric particles **62** and selective gas pressurization to the inner chamber **60**. The plurality of tube windings **58** of the spiral tube are disposed to form a generally cylindrical configuration and bonded together at abutting portions. It is also within the scope of the invention to include a selectively pressurizable elastomeric member (not shown) interposed between the mounting members **52,54** and within the plurality of tube windings **58**. At least one of the mounting members would necessarily include additional valve means for pressurizing the elastomeric member. Preferably, the selectively pressurizable elastomeric member comprises a second spirally configured tube including a second plurality of tube windings disposed within the first spirally configured tube.

In addition, and with specific reference to FIG. **1**, the benefits of the present invention will be realized if the diaphragm **26** is removed from the device illustrated therein. That is, while it is preferred to utilize a diaphragm in the device of the invention, such a diaphragm is not necessary to obtain a device having improved operational characteristics.

It will be readily apparent to those skilled in the art, however, that modification may be made to the structural details of the new flexible wall spring damper as described herein to accommodate particular operational needs and/or requirements. Such changes are not deemed to affect the overall intent or scope of the present invention.

## OPERATION

With particular attention to FIGS. **1-3**, the improved operational characteristics of the new flexible wall spring damper will be specifically discussed.

The invention is a combination spring/damper in that it provides the characteristics of both a spring and a shock absorber in one package. Relative movement between the rigid mounting members **12,14,52,54** of the embodiments of the invention operates to stress the flexible side walls **16,56** and the plurality of elastomeric particles **24** and further operates to vary the volumes of the selectively pressurizable gas chambers **28**. All these actions combine in operation to absorb the shock and/or support the load applied to the spring damper **10,50**. The employment of the elastomeric particles presents a substantial improvement over prior art viscous spring dampers and particularly those viscous spring dampers that employed hydraulic fluids for dampening spring responses to shocks. One particular improved feature of operation is that for a given spring damper size, a significantly longer stroke can be realized with the structure of the present invention as opposed to a structure containing hydraulic fluids. More specifically, the system of a plurality of elastomeric particles may compress where hydraulic fluid will not, thus allowing the present invention to be compressed at a point at which a prior art spring damper would fracture its housing or shear spring or fail at a shear spring bond, thereby releasing fluid. Another operational characteristic and advantage of the present invention is one of improved stability. Since the elastomeric shear springs of a typical prior art viscous spring damper are stressed primarily in shear during operation and shock absorption, instability can occur when the shear spring is stroked too far or "over center". In such an instance, spring tension drops briefly with increased deflection, rather than a steady increase in force on the spring with increased deflection. Once in the position of over center deflection, it is difficult for the spring to rebound or flip back rapidly. Such a slow response and the load versus deflection instability are undesirable characteristics of a viscous spring damper.

With particular reference to FIG. **3**, an operational curve of the inventions of FIGS. **1** and **2** is shown and particularly illustrates that the undesirable operational characteristics of the shear spring being deflected over center are no longer present. The combination of the flexible side wall and elastomeric particles exhibits a steady increase in load versus deflection. Line **76** represents load versus deflection characteristics of the invention upon compression and line **78** exhibits load versus deflection characteristics upon reflection or rebound.

Experimentally, both the compression and rebound lines of the graph of FIG. **3** are determined to be exponential curves. Therefore, it can be shown that the invention exhibits a constant natural frequency characteristic, independent of the mass supported thereby. The importance of having a constant natural frequency spring damper is that when the spring damper is installed on a vehicle, the ride characteristics will remain the same regardless of how the vehicle is loaded.

Where a selectively pressurizable air chamber is included in the invention, the spring rate, and hence the natural frequency of operation of the invention, can be altered by selective pressurizing of the gas chamber. With reference to FIG. **1**, the extent of compression of the spring damper **10** under load can be adjusted by pressure changes in the selectively pressurizable gas chamber **28** or the spring damper chamber **18**. However, the addition of gas to the selectively pressurizable gas chamber **28** results in a much greater increase in the natural frequency and spring rate than does the addition of air to the spring damper chamber **18**. The addition of air to the gas chamber **28** effectively increases the density of the elastomeric particles **24** in the spring damper chamber **18**, since the volume of the spring damper chamber **18** decreases. The addition of air to the spring damper chamber **18** does not change the overall volume of the spring damper chamber **18**; hence, the particle density does not change in this case.

For the spiral dry viscous spring damper (FIG. **2**) the extent of compression at load, spring rate, and natural frequency can be altered by varying the pressure in the inner chamber **60** of connecting member **56** or by varying the pressure in the selectively pressurizable elastomeric member (not shown).

The invention has been described with reference to the preferred and an alternative embodiment. Obviously, modifications and alterations will occur to others upon a reading and understanding of the specification. It is our intention to include all such modifications and alterations insofar as they come within the scope of the appended claims or the equivalents thereof.

Having thus described our invention, we now claim:

1. A flexible wall spring damper including opposed first and second rigid mounting members interconnected by a flexible side wall for defining a flexible wall spring damper chamber, a plurality of solid essentially

incompressible in volume irregular shaped elastomeric particles included in said damper chamber, said plurality of elastomeric particles having particle sizes of 30 mesh or smaller filling said chamber at least to the point wherein under static load said chamber is essentially filled, and a flexible diaphragm connected to said second rigid mounting member, a selectively pressurizable gas chamber is interposed between said second rigid mounting member and said diaphragm, whereby relative movement between said opposed rigid mounting members operates to stress said flexible side wall, vary the volume of said gas chamber, and cause relative interaction between the solid incompressible individual particles thus promoting the desired damping and spring characteristics.

2. The flexible wall spring damper as defined in claim 1 wherein said rigid mounting members include air valves for providing selective gas pressurization to said flexible wall spring damper chamber and said selectively pressurizable gas chamber and said first rigid mounting member includes means for providing elastomeric particles to said flexible side wall chamber and means for filtering the egress of gas from said flexible side wall chamber.

3. The flexible wall spring damper of claim 1 wherein said spring damper chamber further includes an agglomeration inhibitor.

4. The flexible wall spring damper as defined in claim 3 wherein said agglomeration inhibitor is at least one material selected from the group consisting of particulate polytetrafluoroethylene, silica powder and silicone oil.

5. The flexible wall spring damper of claim 1 wherein said spring damper chamber includes said plurality of elastomeric particles sized in aggregation to fill said spring damper chamber upon relative movement between said members to an extent to dispose said members in an equilibrium position upon being subjected to a static load.

6. A spiral dry viscous spring damper including opposed first and second rigid mounting members and a flexible wall connecting member therebetween, said connecting member comprising a first spirally configured tube including a first plurality of tube windings and including a tube inner chamber, a plurality of solid essentially incompressible in volume irregular shaped elastomeric particles are included in said inner chamber, said particles having particle sizes of 30 mesh or smaller substantially filling said tube inner chamber, whereby relative movement between said rigid mounting members operates to stress said spirally configured tube and cause relative interaction between the solid incompressible individual particles thus promoting the desired damping and spring characteristics.

7. The spiral dry viscous spring damper as defined in claim 6 wherein a first end portion and a second end portion of said spiral tube contiguous to said first rigid mounting member and said second rigid mounting member respectively are bonded to said first and said second rigid mounting members and includes port means for supplying said elastomeric particles to said tube and selective gas pressurization to said tube.

8. The spiral dry viscous spring damper as defined in claim 6 wherein said plurality of tube windings of said spiral tube are disposed to form a generally cylindrical configuration and bonded together at abutting portions.

9. The spiral dry viscous spring damper as defined in claim 6 wherein a selectively pressurizable elastomeric member is interposed between said mounting members and within said plurality of tube windings, at least one of said mounting members including valve means for pressurizing said elastomeric member.

10. The spiral dry viscous spring damper as defined in claim 9 wherein said selectively pressurizable elastomeric member comprises a second spirally configured tube including a second plurality of tube windings disposed within said first spirally configured tube.

11. A flexible wall spring damper including opposed first and second rigid mounting members interconnected by a flexible side wall for defining a flexible wall spring damper chamber, a plurality of elastomeric solid essentially incompressible in volume irregular shaped particles included in said damper chamber, said plurality of elastomeric particles having particle sizes of 30 mesh or smaller filling said chamber at least to the point wherein under static load said chamber is essentially filled, whereby relative movement between said opposed rigid mounting members operates to stress said flexible side wall and cause relative interaction between the solid incompressible individual particles thus promoting the desired damping and spring characteristics.

12. The flexible wall spring damper as defined in claim 11 wherein one of said rigid mounting members includes an air valve for providing selective gas pressurization to said flexible wall spring damper chamber, means for providing elastomeric particles to said flexible side wall chamber and means for filtering the egress of gas from said flexible side wall chamber.

13. The flexible wall spring damper of claim 11 wherein said spring damper chamber further includes an agglomeration inhibitor.

14. The flexible wall spring damper as defined in claim 13 wherein said agglomeration inhibitor is at least one material selected from the group consisting of particulate polytetrafluoroethylene, silica powder and silicone oil.

15. The flexible wall spring damper of claim 11 wherein said spring damper chamber includes said plurality of elastomeric particles sized in aggregation to fill said spring damper chamber upon relative movement between said members to an extent to dispose said members in an equilibrium position upon being subjected to a static load.

* * * * *

# United States Patent [19]

**Shtarkman**

[11] **Patent Number: 4,504,044**

[45] **Date of Patent: Mar. 12, 1985**

[54] **DRY VISCOUS SPRING DAMPER**


[75] Inventor: **Emile M. Shtarkman**, Euclid, Ohio

[73] Assignee: **Imperial Clevite Inc.**, Glenview, Ill.



[21] Appl. No.: **436,331**

[22] Filed: **Oct. 25, 1982**

[51] **Int. Cl.**[3] .............................................. **F16F 13/00**

[52] **U.S. Cl.** ....................................... **267/35**; 188/268; 267/63 A; 267/122; 267/140.1; 267/140.3; 267/141; 267/152; 267/153; 280/662; 280/697; 280/712; 280/716

[58] **Field of Search** ............... 188/268; 267/35, 63 A, 267/141, 152, 153, 140.1, 113, 122, 128, 140.3; 280/697, 712, 716, 662



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*Attorney, Agent, or Firm*—Russell E. Baumann


[57] **ABSTRACT**

A dry viscous spring damper is provided particularly adapted for absorbing shocks, dissipating energy and load carrying and leveling of a vehicle. The spring damper is comprised of a first housing member at least partially received in a second housing member and joined at the second housing member with an elastomeric shear spring. The spring damper has a first chamber separated from a second chamber by an elastomeric diaphragm. A valve is provided for selectively pressurizing the second chamber with pressurized gas or air. A plurality of elastomeric particles are included in the first chamber whereby relative movement between the housing members operates to stress the shear spring and the elastomeric particles, and vary the volumes of the first and second chambers.


**15 Claims, 7 Drawing Figures**

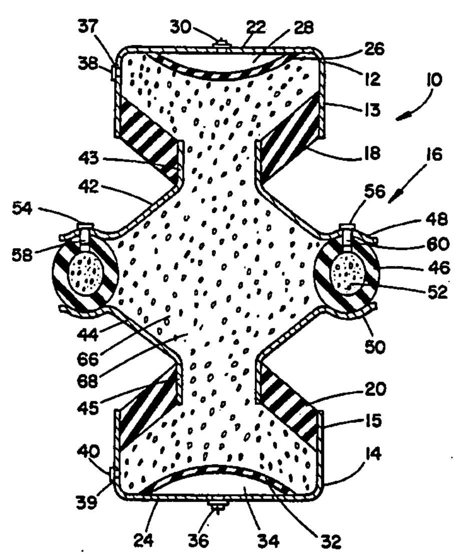

FIG. 1

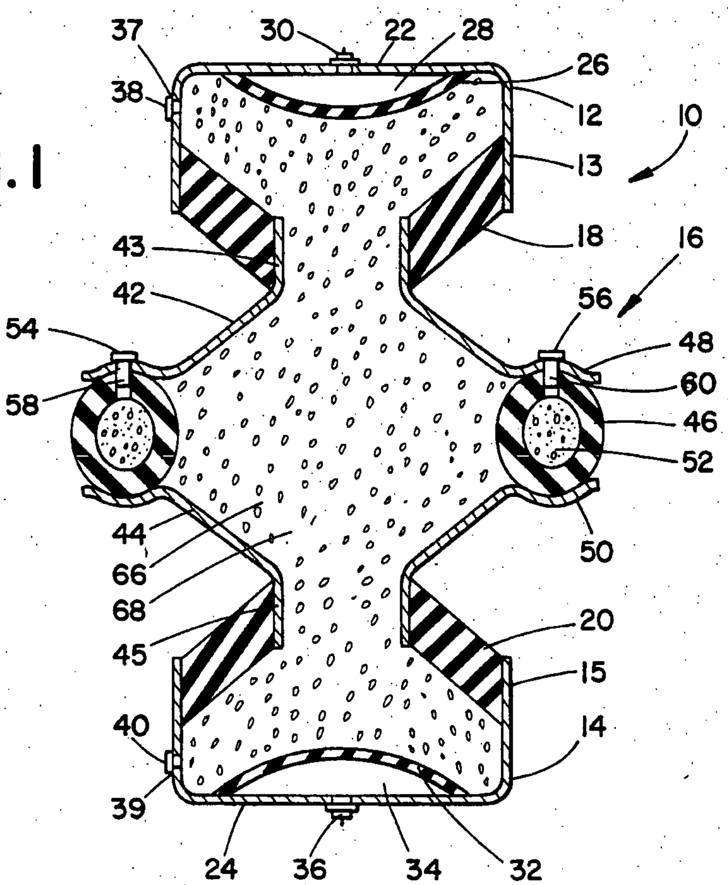
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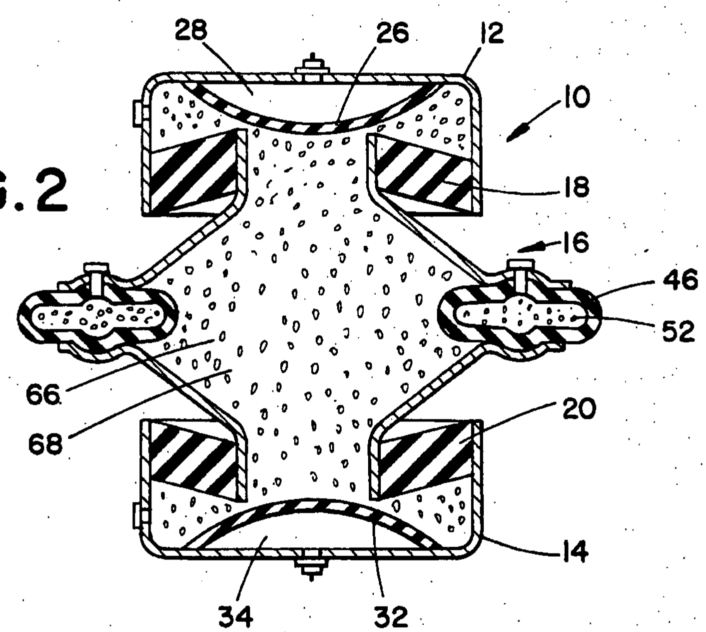
FIG. 2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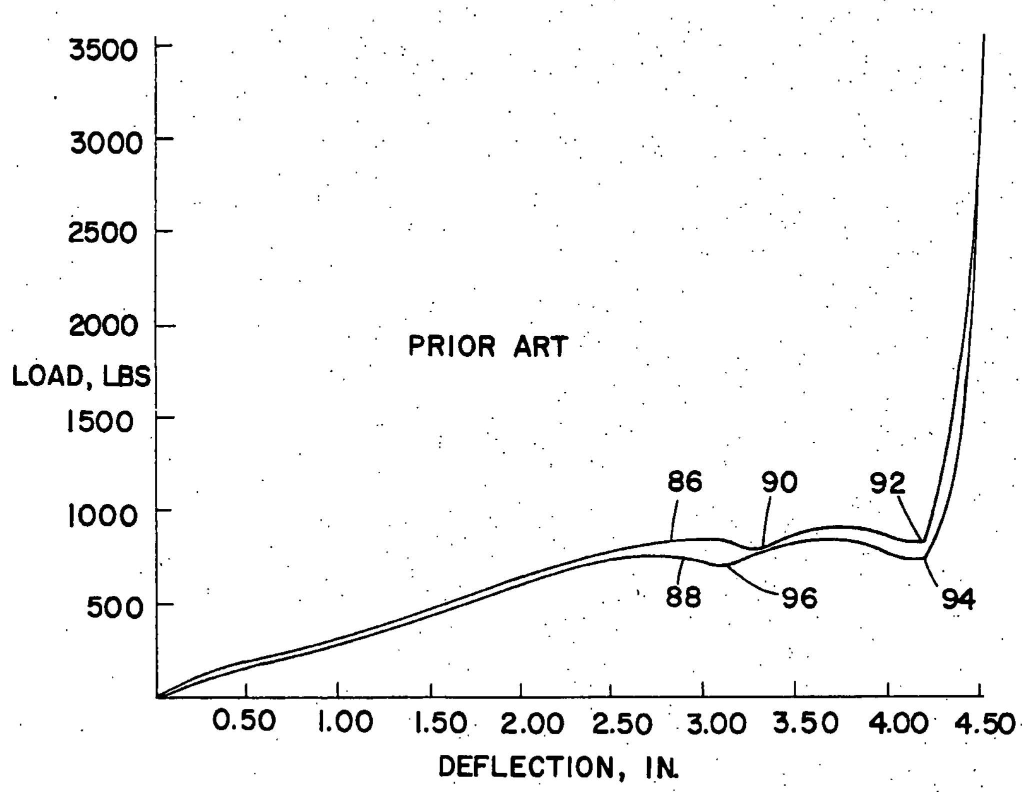

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FIG. 3

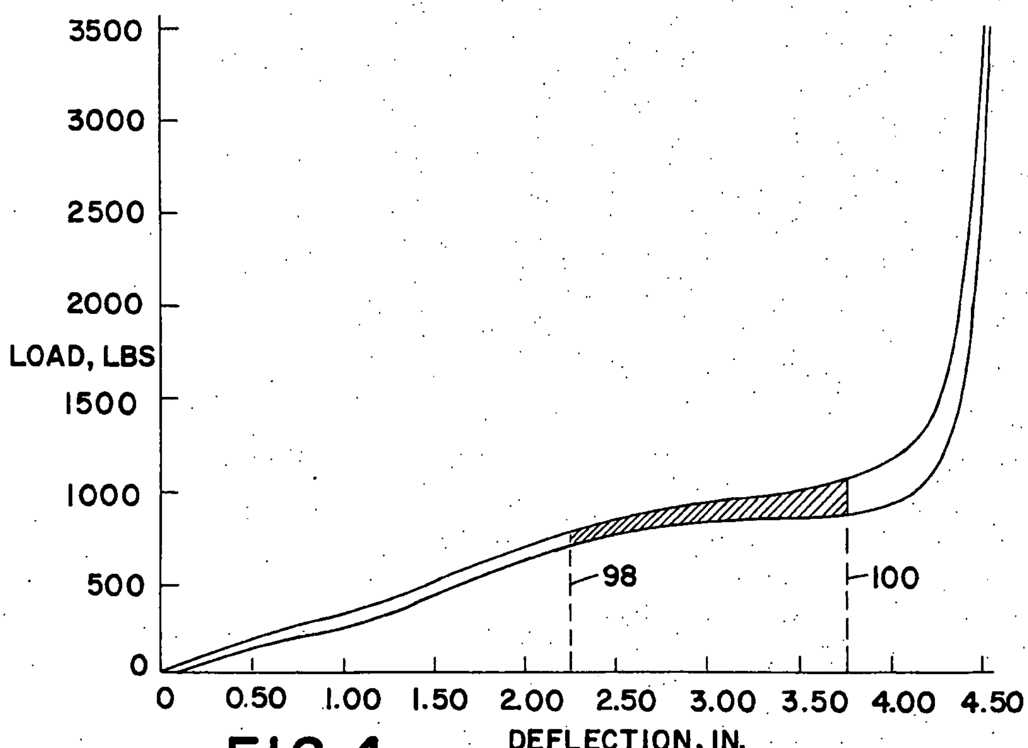

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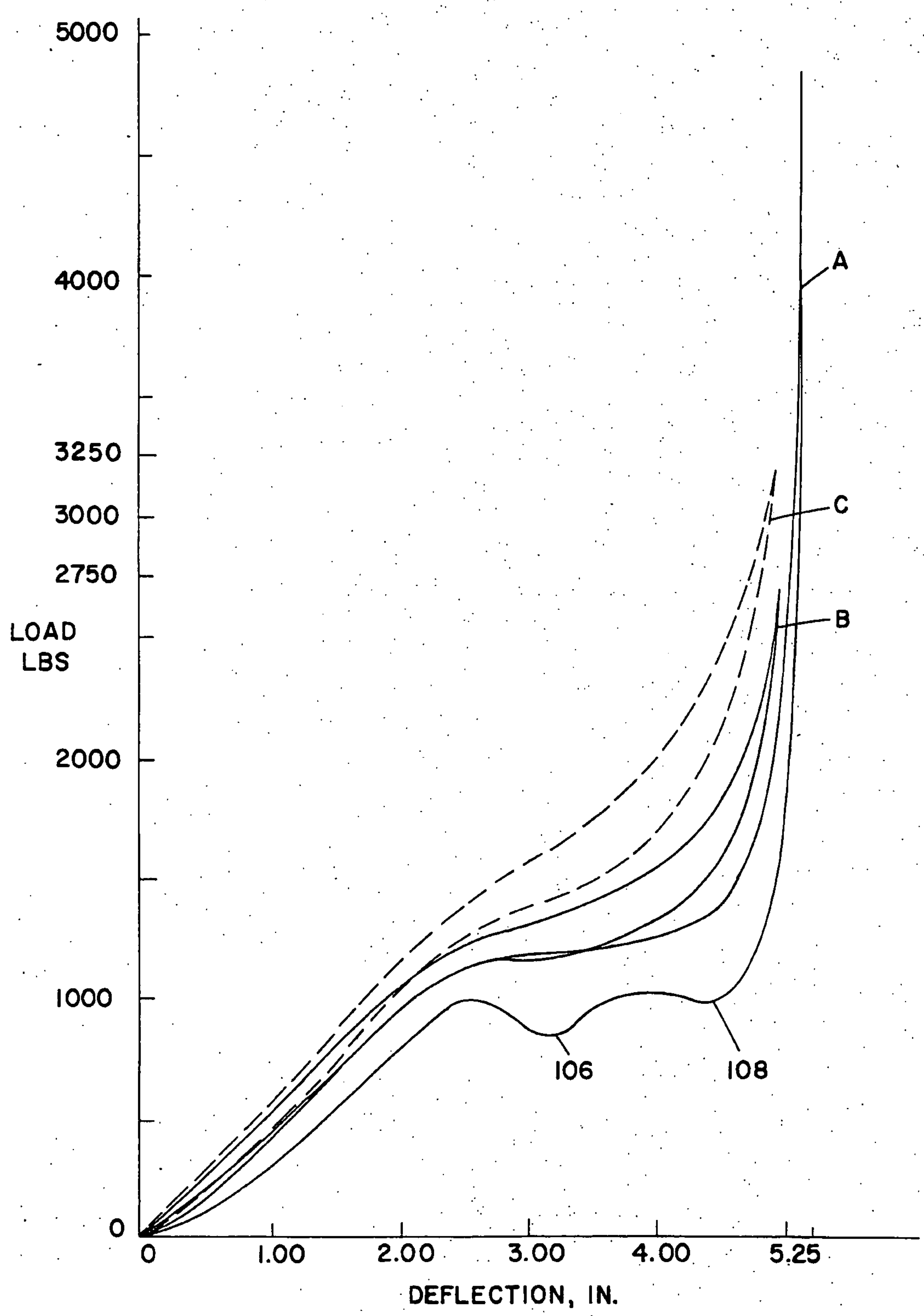
FIG. 4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DEFLECTION, IN.

FIG. 5

FIG. 6

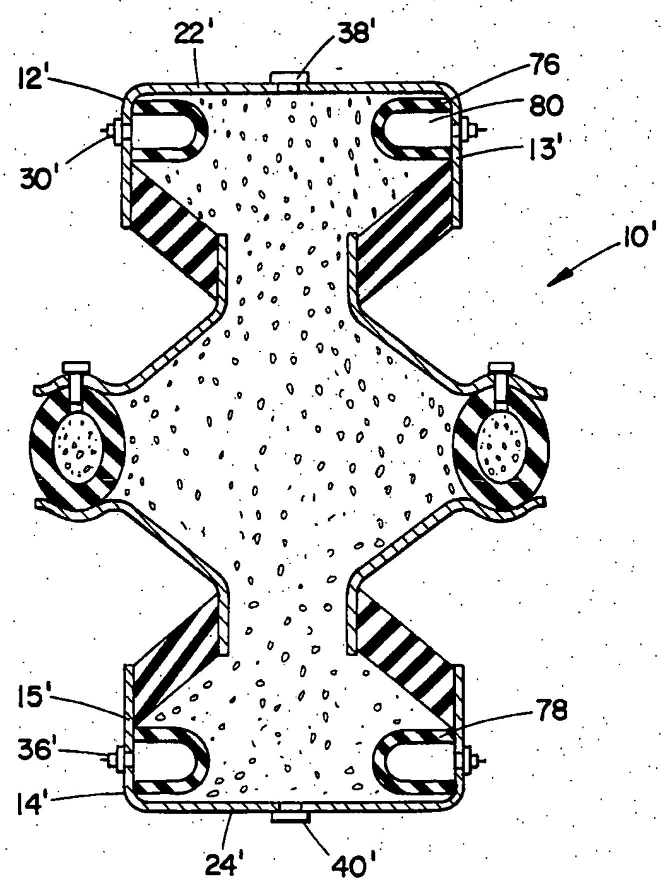
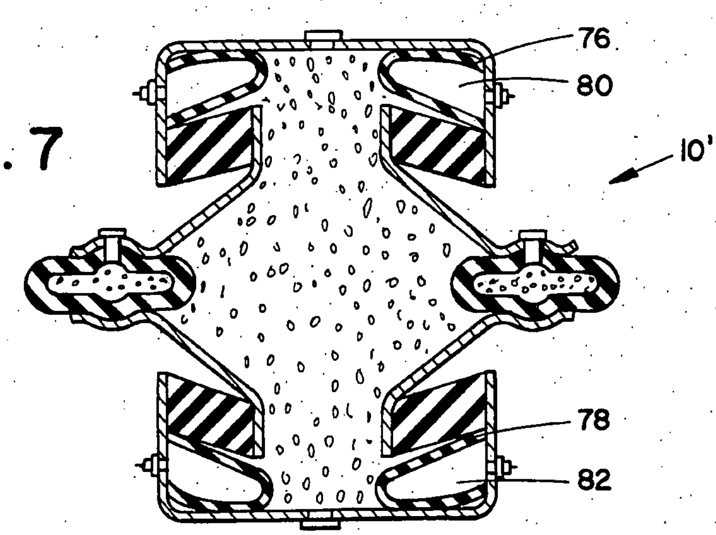



FIG. 7



## DRY VISCOUS SPRING DAMPER

### BACKGROUND OF THE INVENTION

The present invention relates generally to load carrying shock absorbers. More particularly, it relates to shock absorbers of the type which are mounted to vehicles and which use both an elastomeric spring and a selectively pressurizable gas chamber for absorbing shock, energy dissipation and carrying and leveling load. However, it will be appreciated by those skilled in the art that the invention can be readily adapted for use in other environments as, for example, where similar spring dampening devices are employed to protect or cushion other items.

Known spring damper devices of the type described have included elastomeric shear springs, elastomeric diaphragms, selectively pressurizable gas chambers and associated communicating fluid chambers including restrictive orifice means therebetween for retricting the flow of fluid between the fluid chambers. Such a spring dampening device is described in Application Ser. No. 208,013, filed Nov. 18, 1980, now abandoned, in the name of Shtarkman et al and assigned to the assignee of the subject application. In the Shtarkman et al application an expandable and contractable elastomeric bladder for separating a gas chamber from a fluid chamber in a viscous spring damper is provided. Varying the pressurization of the gas chamber is employed for varying the spring rate of the viscous spring damper, calibrating the damper or leveling a load supported thereby. In addition, fluid flow between the communicating fluid chambers absorbs and dampens the shock and spring forces occurring during operation of the device. Such a structure provides the advantageous operating characteristics of both a spring and a shock absorber in one package.

A particular problem inherent in viscous spring dampers including fluid chambers is the limitation of a short stroke. In other words, the extent of compressive reduction of the spring damper device may be unduly limited because the volume of fluid, that must be displaced between communicating fluid chambers, may become too great to allow compressive reduction of the device to a desired level. There may simply not be enough room for the fluid, which is typically an incompressible hydraulic fluid, to be displaced. As a result, failure of the viscous spring damper would occur upon compression of the device beyond a certain level, either occurring through facture of the device's housing or shear spring, or through shear spring bond failure.

Another problem with prior viscous spring dampers is instability of the spring characteristics upon excessive deflection of the device. Specifically, an elastomeric shear spring is stressed primarily in shear upon deflection of a viscous spring damper. However, in situations where the viscous spring damper is stroked or compressed beyond a point where the shear spring becomes unstable, the shear spring operates irregularly. On a load/deflection curve, this is manifested as a drop in load with increasing deflection, rather than a steady increase in load with increasing deflection. Once in this position, it is difficult for the spring to expand or "flip back" rapidly. Such difficulty further results in slow response in the operation of the viscous spring damper.

The present invention contemplates a new and improved dry viscous spring damper which overcomes all of the above referred to problems and others to provide a new viscous spring damper which is simple in design, economical to manufacture, readily adaptable to a plurality of load carrying and shock absorbing uses, and which provides improved shock absorption and energy dissipation.

### BRIEF SUMMARY OF THE INVENTION

In accordance with the present invention, there is provided a dry viscous spring damper for carrying load and dampening structural agitation, comprising a first housing member at least partially received in a second housing member and joined at said second housing member with an elastomeric shear spring, thereby creating a central chamber. A plurality of elastomeric particles having particle sizes of 30 mesh or smaller are included in the first chamber. Relative movement between the outer and inner members operates to stress the shear spring and vary the volume of the chamber. The plurality of elastomeric particles provide spring and dampening forces in combination with the spring and dampening forces of the shear spring.

In accordance with another aspect of the present invention, there is provided a dry viscous spring chamber for carrying load and dampening structural agitation comprising a first housing member at least partially received in a second housing member and joined at said second housing member with an elastomeric shear spring. The spring damper has a first chamber separated from a second chamber by an elastomeric diaphragm and a first valve means for selectively pressurizing the second chamber with pressurized gas. A plurality of elastomeric particles having particle sizes of 30 mesh or smaller are included in the first chamber. Relative movement between the outer and inner members operates to stress the shear spring and vary the volumes of the first and second chambers. The plurality of elastomeric particles provide spring and dampening forces in combination with the spring and dampening forces of the shear spring and diaphragm.

In accordance with still another aspect of the present invention, the plurality of elastomeric particles comprise particles having a preferred aspect ratio of essentially 1 and highly irregular surfaces.

In accordance with a further aspect of the present invention, the first chamber further includes agglomeration inhibitors comprising polytetrafluoroethylene or silica powders in combination with the plurality of elastomeric particles to inhibit agglomeration of the particles during operation of the spring damper.

In accordance with yet another aspect of the present invention, the plurality of elastomeric particles included in the first chamber are sized in aggregation to fill the chamber upon relative movement between the first and second housing members to an extent to deflect the shear spring to a position where the shear spring is disposed over center.

In accordance with the present invention, there is provided a dry viscous spring damper for load carrying and dampening structural agitation comprising a first outer member joined to a connecting member by means of a first elastomeric shear spring and a second outer member joined to the connecting member by means of a second elastomeric shear spring. The first outer member includes a first gas chamber and a first elastomeric diaphragm and the second outer member includes a second gas chamber and a second elastomeric diaphragm. A main chamber is included in the connecting

member and includes a plurality of elastomeric particles whereby relative movement between the outer members and connecting member operates to stress the shear springs, the elastomeric diaphragms and the plurality of elastomeric particles and vary the volumes of the chambers.

In accordance with another aspect of the present invention, the connecting member includes a first rigid wall portion and a second rigid wall portion. The first rigid wall portion is bonded to the first elastomeric shear spring and the second rigid wall portion is bonded to the second elastomeric shear spring. The first rigid wall portion is joined to the second rigid wall portion by a third elastomeric spring. The third elastomeric spring comprises a toroidally configured member including an annular inner chamber, which can include an additional plurality of elastomeric particles, and/or pressurized gas.

In accordance with yet another aspect of the present invention, means are provided for selective gas pressurization of at least one of the elastomeric diaphragms and associated gas chamber whereby the dry viscous spring damper is selectively adjustable to level a load.

One benefit obtained by use of the present invention is an improved viscous spring damper having improved operating characteristics.

Another benefit obtained from the present invention is an improved viscous spring damper having a plurality of elastomeric particles received in a chamber for load carrying, shock absorption, dampening, and energy dissipation.

Another benefit obtained from the present invention is a viscous spring damper which provides significantly longer stroke of operation with improved load versus deflection stability and improved spring response.

Other benefits and advantages for the subject new dry viscous spring damper will become apparent to those skilled in the art upon a reading and understanding of this specification.

## BRIEF DESCRIPTION OF THE DRAWINGS

The invention may take physical form in certain parts and arrangement of parts, the preferred and an alternative embodiment of which will be described in detail in the specification and illustrated in the accompanying drawings which form a part hereof and wherein:

FIG. 1 is a cross-sectional elevational view of a preferred embodiment of a dry viscous spring damper constructed in accordance with the present invention;

FIG. 2 is a cross-sectional view of the dry viscous spring damper of FIG. 1 showing the invention in a compressed state;

FIG. 3 is a graph showing the deflection characteristics of a prior art device;

FIG. 4 is a graph showing the deflection characteristics of the present invention;

FIG. 5 is a graph showing the deflection characteristics of the present invention particularly illustrating the operational characteristics of the invention with modifications to the invention construction;

FIG. 6 is a cross-sectional elevational view of an alternative embodiment of a dry viscous spring damper constructed in accordance with the present invention; and,

FIG. 7 is a cross-sectional view of the dry viscous spring damper of FIG. 6 showing the damper in a compressed state.



## DETAILED DESCRIPTION OF THE INVENTION

Referring now to the drawings wherein the showings are for purposes of illustrating the preferred and an alternative embodiment of the invention only and not for purposes of limiting same, the FIGURES show a dry viscous spring damper **10** particularly useful for installation upon vehicles for carrying and leveling load, absorbing shock and dissipating energy. The invention is a combination of a spring and a damper in that it provides the characteristics of both a spring and a shock absorber in one package.

With particular reference to FIGS. **1** and **2**, a preferred embodiment of a dry viscous spring damper **10** is illustrated. The damper **10** is comprised of a pair of opposed outer rigid members or housings **12,14** which are preferably constructed of a strong and durable material such as steel. First outer housing **12** is connected to an inner housing or connecting member **16** by a first elastomeric shear spring **18**, and second outer member or housing **14** is connected to the inner connecting member **16** by a second elastomeric shear spring **20**. First shear spring **18** is bonded to a side wall portion **13** of first housing member **12** and second shear spring **20** is similarly bonded to a side wall portion **15** of second housing member **14**. First outer housing member **12** includes a first elastomeric diaphragm **26** and a selectively pressurizable gas chamber **28**. Conventional gas fill means such as a valve **30** in base wall portion **22** may be advantageously employed for selective pressurization of the chamber **28**. Similarly, second outer housing member **14** includes a second elastomeric diaphragm **32**, a second selectively pressurizable gas chamber **34** and a conventional gas valve **36** in base wall **24** for filling the chamber **34**. Preferably, air is employed for filling of the gas chambers **28,34**; however, it is within the scope of the invention to include alternate gaseous mediums for filling of the chambers.

Inner housing or connecting member **16** includes a first rigid wall portion **42** bonded to first elastomeric shear spring **18** at flange portion **43** and a second rigid wall portion **44** bonded to second elastomeric shear spring **20** at a flange portion **45**. First rigid wall portion **42** is joined to the second rigid wall portion **44** by a third elastomeric spring **46** bonded to opposed flange portions **48,50** of the first rigid wall portion and the second rigid wall portion, respectively. Third spring **46** preferably comprises a toroidally or donut configured member including an annular inner chamber **52**. Access to inner chamber **52** is provided through filler ports **54,56** which may be typically sealed by conventional plugs **58,60** or by valves.

It is a particular feature of the invention that a main cavity or chamber **66** is provided in the outer housings **12,14** and inner housing **16** between opposed first and second elastomeric diaphragms **26,32** and that such main chamber **66** includes a plurality of solid elastomeric particles **68** to be particularly useful as the working medium for the invention which are essentially incompressible in volume. Elastomeric particles **68** may be constructed of various elastomeric substances, preferably a natural rubber compound with a Shore A harness range of 45 to 70, with an elongation-at-break of at least 500%, and with a carbon black loading such that the hysteretic properties are in the range of those normally used in automotive bushing compounds; and preferably comprise particles sized 30 mesh or smaller. The

particles must be so sized that they act as a system of individual particles. At a particle size of 30 mesh or smaller, this occurs. Such a size range is best suited for employment in a viscous spring damper device of the type of the invention. It is also preferred that the particles have an aspect ratio of essentially 1 and have highly irregular surfaces. Access filler ports **38,40** in outer housing members **12,14** provide for filling main chamber **66** with the particles. These ports **38,40** may be typically sealed by plugs **37,39** or by valves. Additionally, a sealable gas filter means (not shown) may also be included in communication with chamber **66** for the filtered egress of gas or air therefrom.

Maintaining the individuality of the particles **68** is important to the proper functioning of the invention. If the particles become agglomerated, it has been found that performance suffers. Therefore, additives have been utilized to inhibit agglomeration. Such additives tend to coat the particles and not remain free in the main chamber **66**. Particulate additives such as polytetrafluoroethylene powder and various types of silica which coat the elastomeric particles and have a mean particle size much smaller than the elastomeric particles have been found to be particularly useful. Alternatively, a liquid additive, such as silicone oil, may also be employed. In addition, a thin film deposit from a gas could also be utilized.

Because the medium of chamber **66** is a plurality of elastomeric particulates, packing efficiency is such that there are spaces between the individual particles. For non-compacted elastomeric particles with highly irregular surfaces, the particles may occupy less than 25% of the fill volume of chamber **66**. The balance of the volume of chamber **66** may be a vacuum, or be filled with air or other gases such as nitrogen.

The medium of elastomeric particles provides both a spring rate and a dampening capacity for the invention. The complex interactions of the particles which cause deformation of individual particles and movement of the particles relative to each other are what produces these behavior characteristics. The spring and dampening rates can be tailored through appropriate choices of elastomeric particle types, sizes and shapes.

With particular reference to FIG. **1**, the main cavity or chamber **66** of the dry viscous spring damper **10** may be completely filled with elastomeric particles as shown or could be only partially filled. However, when only partially filled, there must be sufficient elastomeric particles present so that the chamber **66** is filled when the device has been compressed to the point where elastomeric shear springs **18,20** extend normally to the bonding flanges **13,15,43,45** of the outer and inner housings. As will be further explained hereinafter, this is the point of deflection where the shear spring is disposed over center, and at which the elastomeric shear springs become unstable. The annular inner chamber **52** of third elastomeric spring **46** may or may not include elastomeric particles, and it may or may not be pressurized with gas.

With particular reference to FIGS. **6** and **7**, an alternative embodiment of the invention is shown. For ease of illustration and appreciation of this alternative, like components are identified by like numerals with a primed (') suffix, and new components are identified by new numerals. The alternative embodiment of a dry viscous spring damper **10'** includes an alternately configured first elastomeric diaphragm **76** and selectively pressurizable gas chamber **80** and a second elastomeric diaphragm **78** and gas chamber **82** received in outer members or housings **12',14'**. The end portions of the elastomeric diaphragms **76,78** are bonded against the side walls **13',15'** of outer members **12',14'** rather than against the base walls **22',24'** as in the preferred embodiment. Such a configuration requires associated movement of the gas valves **30',36'** to the side walls of the outer housings **12',14'**. Filler ports **38',40'** may then be located in the base walls of the outer housings.



In addition, and with reference to FIGS. **1** and **6**, the benefits of the present invention will be achieved if diaphragms **26,32** and **76,78**, respectively, are removed from the device illustrated therein. That is, while it is preferred to utilize a diaphragm in the device of the invention, such a diaphragm is not necessary to obtain a device having improved operational characteristics.

It will be readily apparent to those skilled in the art, however, that modification may be made to the structural details of the new dry viscous spring damper as described herein to accommodate particular operational needs and/or requirements. Such changes are not deemed to effect the overall intent or scope of the present invention. A structural including a first housing member joined to a second housing member by an elastomeric shear spring and including at least one elastomeric diaphragm and further including a selectively pressurizable gas chamber and a main cavity for receiving a plurality of elastomeric particles between the housing members and selectively pressurizable gas chamber would fall within the scope of the present invention.

## OPERATION

With particular attention to FIGS. **2** through **5** and **7**, the improved operational characteristics of the new dry viscous spring damper will be specifically discussed.

The invention is a combinatiom spring/damper in that it provides the characteristics of both a spring and a shock absorber in one package. FIG. **2** illustrates the device of FIG. **1** in a compressed state where elastomeric shear springs **18,20** have been stressed; where elastomeric diaphragms **26,32**, pressurized gas chambers **28,34**, connecting member elastomeric spring **46** and inner annular chamber **52** are compressed; and where the plurality of elastomeric particles **68** contained in the main cavity or chamber **66** and annular inner chamber **52** are also compressed. FIG. **7** illustrates the device of FIG. **6** in a compressed state where alternative elastomeric diaphragms **76,78** have been stressed and gas chambers **80,82** have been compressed. All these actions combine in operation to absorb the shock and/or support the load applied to the viscous spring damper **10**. The plurality of elastomeric particles **68** further operates to dampen the spring response to the shock or load and to dissipate the energy of the spring upon expansion of the springs **18,20,46** to an equilibrium state. The employment of the elastomeric particles **68** presents a substantial improvement over prior art viscous spring dampers and particularly those viscous spring dampers that employed hydraulic fluids for dampening spring responses to shocks. One particular improved feature of operation is that for a given spring damper size, a significantly longer stroke can be realized with the structure of the present invention as opposed to a structure containing hydraulic fluids. More specifically, the system of a plurality of elastomeric particles **68** may compress where hydraulic fluid will not, thus allowing the present invention to be com-

pressed to a point at which a prior art spring damper would fracture its housing or shear spring, or fail at a shear spring bond, thereby releasing fluid.

Another operational characteristic and advantage of the present invention is one of improved stability. Since the elastomeric shear springs of a viscous spring damper are stressed primarily in shear during operation and shock absorption, instability can occur when the shear spring is stroked too far or "over center". In such an instance, spring tension drops briefly with increased deflection. With particular reference to FIG. 3, a load/deflection curve of a spring damper similar in construction to that of FIG. 1 but with appropriate restrictive orifice and hydraulic fluid flowing between communicating fluid chambers instead of elastomeric particles as illustrated. One line **86**, of the curve of FIG. 2 represents the load/deflection curve on the compression of the device and a second line **88**, represents the load deflection curve on rebound or reflection. It may be seen that two points of drop **90,92** and **94,96** occur respectively in the curves. The drops illustrate the point when the elastomeric shear springs are deflected over center and exhibit a drop in load with increasing deflection, rather than a steady increase in load on the spring with deflection. Once in the position of a drop or over center deflection, it is difficult for the spring to rebound or flip back rapidly. Such a slow response and the load versus deflection instability are undesirable characteristics of a viscous spring damper.

With particular reference to FIG. 4, an operational curve of the invention of FIG. 1 is shown and particularly illustrates that the undesirable operational characteristics of the shear spring being deflected over center are no longer present. The invention includes sufficient aggregate amounts of elastomeric particles such that the elastomeric particles are at least beginning to be compessed at a point on the curve before the drops or instability occur. The forces required to compress the elastomeric particles are then sufficient to overcome the loss of load which occurs when the elastomeric shear springs go over center, and as illustrated in the drops **90,92** and **94,96** of FIG. 3. With the plurality of elastomeric particles **68** in the main cavity or chamber **66** being under compression when the shear springs go over center, there is a substantial restoring force present to urge the shear springs back into their normal equilibrium position when the compressive load is removed. This provides the improved response time of the present invention. With reference to FIG. 4, it may be seen that the resulting load/deflection curve for the invention exhibits a steady increase in load versus deflection. In addition, such steady increase is a relatively flat sloped increase between points **98,100** which are the normally expected operating ranges of an automobile. Such a flat slope indicates a reasonably low spring rate for the viscous spring damper, which translates into a soft ride for rider comfort.

A means of load leveling can be achieved by pressurizing the main chamber **66**. That is, if the device were normally statically loaded so as to be compressed a certain amount, when the static load is increased, the device is compressed a greater amount. To enable the device with a heavier load to have the same displacement as a normally loaded device, air can be added to the main chamber until the desired displacement is achieved. This is a desirable feature for an automobile rear suspension, which can experience substantial variation in load depending on passenger and trunk loading.

The present invention also provides alternate load leveling ability, and improved operational flexibility by the addition of selectively pressurizable gas chambers **28,34** and the toroidal or donut configured third elastomeric spring **46**. As opposed to the addition of air or gas to the main chamber, the addition of air or gas to the selectively pressurizable gas chambers effectively increases the density of the elastomeric particles **68** contained in the main or central cavity **66** since the volume of the central cavity decreases upon increase in volume of the selectively pressurizable gas chambers **28,34**. The higher particle density results in a greater spring rate for the device and a faster response for the device in rebound or reflection.

With particular reference to FIG. 5, curve A illustrates the load/deflection curve for a device similar to the construction of the device of FIG. 1 but lacking selectively pressurizable gas chambers **26,34** and with the main cavity **66** substantially filled with elastomeric particles. In the rebound portion of curve A, two drops **106,108** occur and illustrate that the elastomeric shear springs rebound or "jump back" faster than the aggregate of elastomeric particles can follow. Curves B and C of FIG. 5 are for the same device with approximately half the amount of elastomeric particles in the main cavity **66** but with selectively pressurizable air chambers **28,34** pressurized with air. Curve C indicates the invention including air chambers pressurized approximately three times the pressurization of curve B. Curves B and C illustrate that the invention can respond more rapidly with pressurization of gas chambers **26,34**, despite the reduced amount of elastomeric particles, as shown by the lack of drops in the rebound curves. Also, the increase in slope between curves B and C shows how the spring rate increases as a function of increase in air pressure in the gas chambers. It has been found that as the elastomeric particle density remains fixed by the addition of air to the central cavity, the response time in rebound for the invention does not change and the spring rate in the operating region of the invention does not increase as rapidly as a function of pressure as for the case where the pressure is increased in the selectively pressurizable gas chambers. While the two drops **106,108** of curve A can be eliminated by the addition of more elastomeric particles to the main chamber, usually this will increase the spring rate to an unacceptable level, and would limit the maximum eflection possible with the device, which is one of its most important characteristics.

The toroidally configured or donut shaped elastomeric spring **46** of connecting member **16** also provides improved operational flexibility. Spring **46** serves as a soft, flexible connection between the rigid wall portions **42,44** of connecting member **16**. In any type of vehicle suspension, some degree of pivoting is involved as wheels traverse bumps and dips. Depending on how and where the spring and shock absorber system of the suspension is mounted, a certain amount of bending or twisting of the suspension system is required. The donut configured elastomeric spring **46** of the present invention provides means for the invention to easily bend about its middle portions and thereby operates to preserve column loading on the elastomeric shear spring **18,20** which, in turn, provides improved operation. The aspect of the invention that the donut shaped spring **46** may include elastomeric particles or pressurized gas in its inner annular chamber **52** has also been found to provide improved spring and dampening operation.

The invention has been described with reference to the preferred and an alternative embodiment. Obviously, modifications and alterations will occur to others upon the reading and understanding of the specification. It is out intention to include all such modifications and alterations insofar as they come within the scope of the appended claims or the equivalents thereof.

Having thus described out invention, we now claim:

1. A viscous spring damper for carrying and leveling load and dampening structural agitation comprising a first housing member at least partially received in a second housing member and joined to said second housing member with an elastomeric shear spring, said spring damper having a first chamber separated from a second chamber by an elastomeric diaphragm, a first valve means foe selectively pressurizing said second chamber with pressurized gas, and a plurality of solid essentially incompressible in volume irregularly shaped elastomeric particles having particle sizes of about 30 mesh or smaller included in said first chamber and filling said first chamber at least to the extent that the particles interact one with another prior to the point of said shear spring being disposed over center, whereby relative movement between said members operates to stress said shear spring and vary the volumes of said first and second chambers thereby causing relative interaction between the solid incompressible individual particles thus promoting the desired damping and spring characteristic.

2. The viscous spring damper of claim 1 wherein said first chamber is provided with sealable gas filter means, and said plurality of elastomeric particles have an aspect ratio of about 1 and have highly irregular surfaces.

3. The viscous spring damper of claim 1 wherein said first chamber further includes an agglomeration inhibitor.

4. The viscous spring damper of claim 3 wherein said agglomeration inhibitor is at least one material selected from the group consisting of particulate polytetrafluoroethylene, silica particles and silicone oil.

5. In a viscous spring damper including inner and outer members connected by an elastomeric shear spring, having a first and a second chamber therebetween and an expandable and contractable gas chargeable elastomeric diaphragm segregating said first chamber from said second chamber, and a first valve means for selectively charging said diaphragm with gas pressure, the improvement comprising:

a plurality of solid essentially incompressible in volume irregular shaped elastomeric particles received in said first chamber and filling said first chamber at least to the extent that the particles interact one with another prior to the point of said shear sprinhg being disposed over center, said particles having a particle size of about 30 mesh or smaller and wherein relative movement between said member causes relative interaction between the solid incompressible individual particles thus promoting the desired damping and spring characteristics.

6. A viscous spring damper for carrying load and dampening structural agitation comprising a first housing member at least partially received in a second housing member and joined to said second housing member with an elastomeric shear spring, thereby creating a chamber, and a plurality of solid essentially incompressible in volume irregular shaped elastomeric particles having particle sizes of about 30 mesh or smaller included in said chamber and filling said chamber at least to the extent that the particles interact one with another prior to the point of said shear spring being disposed over center, whereby relative movement between said members operates to stress said shear spring and vary the volume of said chamber thereby causing relative interaction between the solid incompressible individual particles thus promoting the desired damping and spring characteristics.

7. The viscous spring damper of claim 6 wherein said chamber is provided with a sealable gas filter means and said plurality of elastomeric particles have an aspect ratio of about 1 and have highly irregular surfaces.

8. The viscous spring damper of claim 6 wherein said chamber includes an agglomeration inhibitor.

9. The viscous spring damper of claim 8 wherein said agglomeration inhibitor is at least one material selected from the group consisting of particulate polytetrafluoroethylene, silica particles and silicone oil.

10. A dry viscous spring damper for carrying load and dampening structural agitation comprising:

a first outer member joined to a connecting member by means of a first elastomeric shear spring, said first outer member including a first gas chamber and a first elastomeric diaphragm;

a second outer member joined to said connecting member by means of a second elastomeric shear spring, said second outer member including a second gas chamber and a second elastomeric diaphragm; and,

a main chamber included in said connecting member containing a plurality of solid essentially incompessible in volume irregular shaped elastomeric particles which fill said main chamber at least to the extent that the particles interact one with another prior to the point of said shear springs being disposed over center having a particle size of 30 mesh or smaller whereby relative movement between said members operates to stress said shear springs, said elastomeric diaphragms, and said plurality of elastomeric particles and vary the volumes of said chambers whereby relative interaction between the solid incompressible individual particles promotes the desired damping and spring characteristics.

11. The dry viscous spring damper as defined in claim 10 wherein said connecting member includes a first rigid wall portion and a second rigid wall portion, said first rigid wall portion being joined to said first elastomeric shear spring and said second rigid wall portion being joined to said second elastomeric shear spring, said first rigid wall portion being joined to said second rigid wall portion by a third elastomeric spring.

12. The dry viscous spring damper as defined in claim 11 wherein said third elastomeric spring comprises a toroidally configured member including an annular inner chamber.

13. The dry viscous spring damper as defined in claim 12 wherein said annular inner chamber includes an additional plurality of elastomeric particles.

14. The dry viscous spring damper defined in claim 13 wherein a sealable access port is provided through said rigid wall portion of said connecting member and through said toriodally configured elastomeric member for the supply of said plurality of elastomeric particles and selective gas pressurization of said annularly configured chamber.

15. The dry viscous spring damper as defined in claim 10 wherein at least one of said outer members includes means for selective gas pressurization of at least one of said elastomeric diaphragms whereby said dry viscous spring damper is selectively adjustable to level a load supported thereby by pressurization of said diaphragm.

* * * * *

# United States Patent [19]

Chung

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[11] 3,929,729

[45] Dec. 30, 1975


[54] POLYUREAURETHANE SHOCK ABSORBING UNIT


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[73] Assignee: The Goodyear Tire & Rubber Company, Akron, Ohio



[22] Filed: Nov. 21, 1974

[21] Appl. No.: 525,790

[52] U.S. Cl. ................ 260/75 NH; 213/7; 248/358; 260/77.5 AM; 260/77.5 AN; 264/236; 267/63; 267/138

[51] Int. Cl.$^2$.. C08G 8/28; B61G 11/08; F16F 7/12; F16F 1/36

[58] Field of Search ... 260/75 NH, 77.5 AM; 213/7; 248/358; 264/236; 267/138, 63



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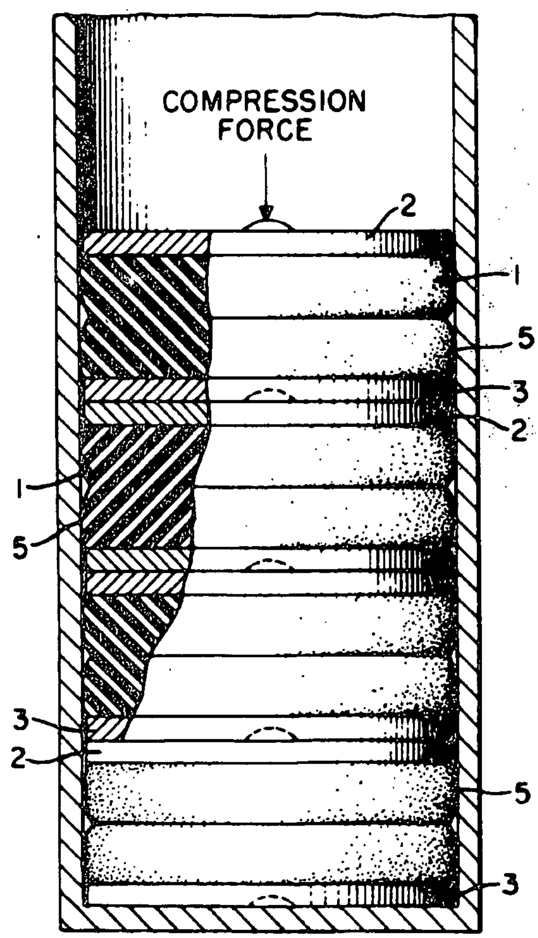
[57] ABSTRACT

Shock absorbing unit which comprises a shaped resilient polyureaurethane composition characterized by being able to withstand compressive cycling and particularly being characterized by retaining a high degree of rebound after a multitude of compressive cycling cycles. Said unit can be suitable for use in a railroad draft gear. Such polyureaurethane composition for said shock absorbing unit is prepared by reacting specifically selected diamines with a corresponding balance of selected diisocyanates in combination with mixtures of certain polymeric polyols with a manipulation of their molecular weights.

12 Claims, 3 Drawing Figures

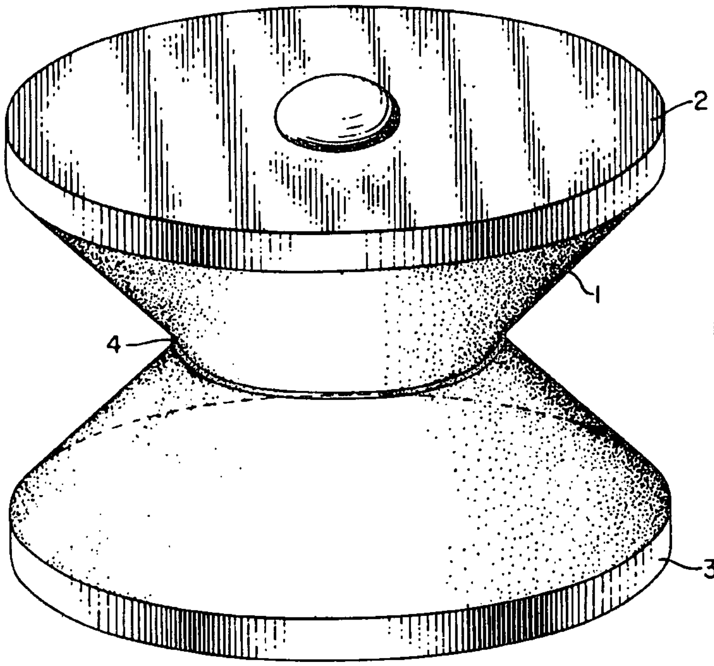



FIG. 1

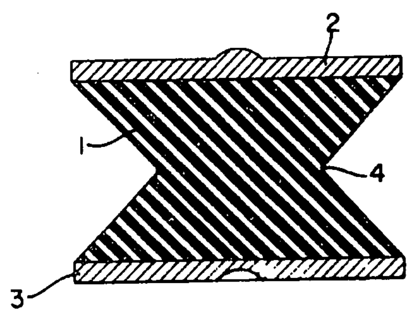



FIG. 2

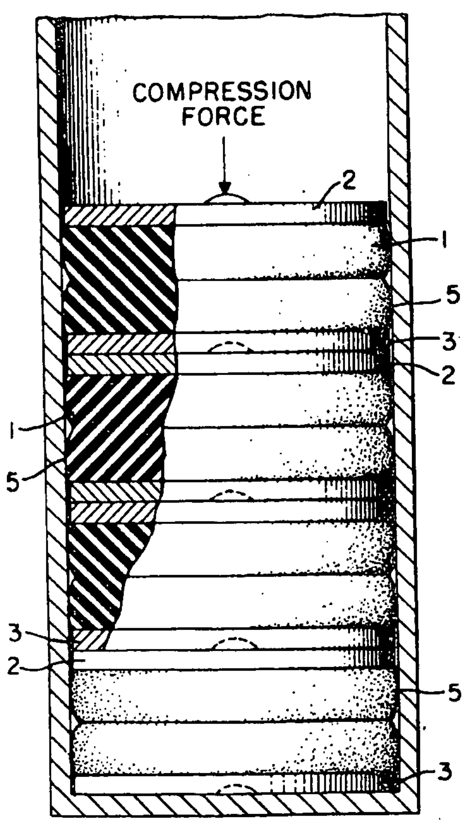



FIG. 3

# POLYUREAURETHANE SHOCK ABSORBING UNIT

This invention relates to shock absorbing units having dynamic shock absorbing ability. The invention more particularly relates to shock absorbing elements for railroad car draft gears of resilient cured polyureaurethane compositions which can resist softening after dynamic compressive cycling under constant compression and to a method of their preparation.

Railroad draft gear shock absorbing units have undergone an evolutionary state of development. Because of the extremely large and repetitive shocks which the very small absorbing units must be able to dynamically withstand, a combination of dynamic compressive test and shock absorbing element composition have had to be coordinated. Indeed, the shock absorbing material must be extremely resistant to softening under repetitive shock loads. A softened shock absorber doesn't absorb enough shock. A simple substitution of materials has been found to be ineffective. As dynamic compressive endurance evaluation procedures and requirements have become more exacting, the shock absorbing element composition is required to become more sophisticated.

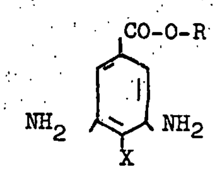
Of particular importance is resistance to softening under demanding requirements of a dynamic compressive cycling test as a measure for determining suitable units for railroad draft gear service. This test requirement is in addition to typical deflection, low temperature and drop hammer tests involving visual destruction of a unit. Indeed, the softening test is more subtle, since a unit can soften without visual signs of disintegration. The softening determination aspect of a compressive cycling test can be exemplified by obtaining a resilient shock absorbing element comprising a resilient composition of a short cylindrical shape measuring about 6.5 inches diameter and about 1.5 inches high with its sides in the shape of a concave V and with its ends covered and adhered to the face of circular steel plates and cycling said element under substantially constant compression alternating between a maximum of about 45 to about 55 percent of its original uncompressed polyureaurethane element height and a minimum of about 8 to about 12 percent of its said original height. Thus, the unit is always under some degree of compression with about 55 percent being the maximum. The force necessary to achieve the maximum compression (about 55 percent) is measured.

The polyureaurethane element itself constantly undergoes a very substantial change in shape as it is compressed and decompressed during each dynamic cycle. Under this relatively severe test, a typical unit can break down or crack within about 50 to about 100 cycles. However, a suitable unit for railroad draft gear service should last or withstand at least about 500 cycles.

A suitable unit for railroad draft gear service should maintain its compression resistance, or resistance to softening, by requiring at least about 4,200 pounds per square inch to compress the unit 55 percent of its original uncompressed height after about 500 cycles of the compressive cycling test.

Therefore, in view of these substantial and demanding shock absorbing compressive endurance requirements, it is an object of this invention to provide an improved resilient shock absorbing unit.

In accordance with this invention, it has been discovered that an improved shock absorbing unit comprises a shaped resilient polyureaurethane composition characterized by (A) withstanding compressive cycling for at least about 500 cycles under constant compression alternating between a maximum of about 45 to about 55 percent and a minimum of about 8 to about 12 percent of its original uncompressed height, (B) requiring at least about 4,200 pounds per square inch to compress said unit 55 percent of its original uncompressed height after 500 cycles of said compressive cycling, and (C) deflecting from about 0.3 to about 0.6 inch, preferably from about 0.4 to about 0.5 inch, at about 25°C upon the application of about 1800 pounds per square inch uniformly to the end surface areas of the said resilient compositions in its uncompressed state when the said composition is a generally disc-shaped cylindrical element with circular parallel end surfaces, said surfaces covered and adhered to circular steel plates, having a diameter of about 1.5 inches and a sidewall connecting the end surfaces substantially in the form of a V-shaped groove having substantially equal length sides, the said groove extending between the said end surfaces, the volume of the solid portion of said element being about 150 to about 170 percent, preferably about 150 percent, of the volume of the said groove, where said resilient shock absorbing polyureaurethane composition is prepared by the method which comprises reacting a diamine selected from 2,2'-diaminodiphenyldisulfide, 2,5-dichloro-1,3-phenylene diamine and a substituted diamine of the formula



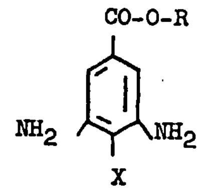
where R is selected from saturated alkyl radicals of 1 to 20 carbon atoms, aryl radicals of 6 to 10 carbon atoms, aralkyl radicals of 7 to 10 carbon atoms and cycloalkyl radicals of 5 to 10 carbon atoms, and X is a halogen radical, with the reaction product of (A) a diisocyanate selected from 1,5-naphthalene diisocyanate when the diamine is selected from 2,2'-diaminodiphenyldisulfide or a diisocyanate selected from 1,5-naphthalene diisocyanate, 3,3'-bitolyene-4,4'-diisocyanate and 3,3'-dimethyldiphenylenemethane-4,4'-diisocyanate when the diamine is selected from said substituted diamine and 2,5-dichloro-1,3-phenylene diamine, and (B) a mixture of at least one polymeric polyol with a total average molecular weight of about 1500 to about 2100 which comprises (1) polyols having a molecular weight of about 1800 to about 2200 selected from (a) about 65 to about 100 weight percent of a polyether polyol or (b) about 65 to about 100 weight percent of a polyol mixture comprising (i) about 35 to about 65 weight percent polyether polyol and (ii) about 65 to about 35 weight percent polyester polyol or (c) about 65 to about 95 weight percent of said polyether polyol-polyester polyol mixture and correspondingly (2) about 35 to about zero or 5 weight percent of at least one of a polyether polyol and polyester polyol having a molecular weight of about 800 to about 1250, wherein said polyether polyol is selected from polytetramethylene ether glycol and polypropylene ether glycol, and

said polyester polyol is selected from (i) caprolactone polyesters prepared from caprolactones containing 6 to 8 carbon atoms and glycols containing 4 to 7 carbon atoms and (ii) azelates of azelaic acid and glycols containing 4 to 7 carbon atoms, where the ratio of isocyanato groups to the sum of hydroxyl groups of the polyols is about 1.7 to about 2.5, where the ratio of primary amino groups of the diamine to excess isocyanato groups over the sum of said hydroxyl groups is about 0.6 to about 1.1 and where the acid number of the polyols is less than about 1.

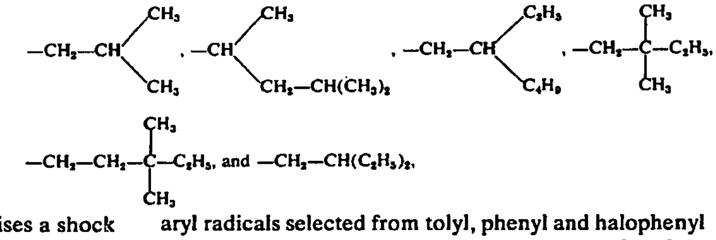
Thus a shock absorbing unit of this invention, suitable for use in a railroad draft gear, comprises a shock absorbing element, the said element comprising a solid disc-shaped resilient polyureaurethane composition of this invention having the said characteristic compression endurance, said resistance to softening and said deflection characteristics at about 25°C and having two opposing and substantially parallel force-receiving surfaces connected by at least one sidewall, preferably a concave sidewall, and having rigid force-receiving plates, preferably metal plates, adhered to its force-receiving surfaces. Correspondingly, the shock absorbing device of a railroad draft gear comprises a series of such units, such as about 8 to about 12 and preferably 10, loaded in a cylinder in series to the shock load, with their force-receiving plates facing against each other.

The resilient polyureaurethane compositions of this invention can be further characterized by a −40°C cold temperature compression test in addition to the deflection characterization test at about 25°C. According to this test, the said composition at about −40°C when shaped and cured to a solid circular disc with a straight sidewall having a diameter of about 1.13 inch and a thickness of about 0.5 inch, requires a maximum pressure of 7500, and preferably a maximum pressure of 6000, pounds per square inch, applied to its flat surfaces to compress the disc 40 percent. This cold temperature compression test is a measure of stiffening of the polyureaurethane composition at low temperatures. It is a measure of the composition's ability to absorb energy without hardening and transmitting shock directly without absorption. A shock absorber of this invention has essentially bottomed out when its percent compression is substantially constant and its percent compression versus load curve substantially horizontal at high load values. The quality of high energy absorption without bottoming out is particularly required for railroad draft gears which are subject to large shocks over a relatively wide range of temperatures including temperatures down to about −40°C.

In the practice of this invention, for the prescribed diamine having the formula



representative examples of R are alkyl radicals selected from methyl, ethyl, n-propyl, isopropyl, n-butyl, isobutyl, n-pentyl, iso-pentyl, hexyl, heptyl, octyl and decyl radicals as well as branched chain radicals such as



aryl radicals selected from tolyl, phenyl and halophenyl radicals, aralkyl radicals selected from benzyl and α-alkyl benzyl radicals, where the alkyl group has 1 to 6 carbon atoms, and cycloalkyl radicals selected from cyclopentyl, cyclohexyl, 4-t-butylcyclohexyl and cycloheptyl radicals.

Preferably, R is selected from alkyl radicals having 1 to 6 carbon atoms.

Representative of X are chlorine, bromine, iodine and fluorine radicals, of which chlorine is preferred.

Representative of various diamines are (methyl)-4-bromo-3,5-diaminobenzoate, (methyl)-4-iodo-3,5-diaminobenzoate, (ethyl)-4-fluoro-3,5-diaminobenzoate and (2-ethylpropyl)-4-chloro-3,5-diaminobenzoate. The (2-ethylpropyl)-4-chloro-3,5-diaminobenzoate is preferred.

In the specification the term "total average molecular weight of about 900 to about 1500" is used to describe the total resulting molecular weight equivalent of a mixture of the polyether polyols and the mixture of polyether polyols and polyester polyols. Thus, such a mixture having a total average molecular weight of 1400 can consist of individual polyols having, for example, molecular weights of 1000, 1250 and 2000. Also, for example, a polytetramethylene ether glycol with an average molecular weight of 1500 mixed with a polyester polyol having a molecular weight of 1000, can be the molecular weight equivalent of mixing the polyester polyol having its individual average molecular weight of 1000 with two polytetramethylene ether glycols having individual average molecular weights of about 1000 to about 2000.

Preferably, the polyether polyol-polyester polyol mixture is selected from (a) polytetramethylene ether glycols having an average molecular weight of about 900 to about 1100 or a mixture with molecular weights of about 900 to about 1100 and of about 1900 to about 2100 and (b) at least one of the polyester polyols selected from hexane diol adipate and azelate having molecular weights of about 800 to about 1200, tetramethylene azelate having a molecular weight of about 1800 to about 2200 and polyesters of ε-caprolactone and diethylene glycol having molecular weights of about 1100 to about 1400 and about 1800 to about 2200.

It is a critical feature of this invention that the ratio of isocyanato groups of the diisocyanate to the sum of the

hydroxyl groups of the polyether polyol (polytetramethylene ether glycol) and polyester polyol is from about 1.7 to about 2.5 and preferably from about 1.8 to about 2.2.

It is preferred that a sufficient amount of diamine is used to provide a ratio of primary amino groups to excess isocyanato groups of the diisocyanate over the sum of the hydroxyl groups of the polyether polyols and polyester polyols (reactive hydrogen-containing materials) in the range of about 0.6 to about 1 and preferably from about 0.7 to about 0.95. Thus, for example, it is desired that from about 0.4 to about 1.1 moles of the diamine is added to the reaction product of the mixtures comprising correspondingly from about 1.7 to about 2.5 moles, preferably from about 1.8 to about 2.0 moles, of the diisocyanate and about 1 mole of the polyether polyol or mixture of polyether polyol and polyester polyol.

It is a further required feature of the invention that the polytetramethylene ether glycols, the caprolactone polyesters, the adipates, the azelates and their mixture have an acid number of less than about 1, desirably less than about 0.5 and more preferably less than about 0.1.

The polytetramethylene ether glycol is of the structure and composition typically prepared from tetrahydrofuran with the aid of an alkylene oxide initiator having 2 to 4 carbon atoms, such as ethylene oxide.

The caprolactone polyesters are substantially linear, hydroxyl-terminated polymers prepared by reacting a caprolactone having 6 to about 8 carbon atoms in the ring, preferably 6 carbon atoms, with a glycol having 4 to 7 carbon atoms and preferably 4 to 6 carbons. Various suitable caprolactones include ε-caprolactone, zetacaprolactone and eta-caprolactone. Alkyl substituted caprolactones can be used with alkyl substituents containing 1 to 2 carbon atoms selected from methyl and ethyl radicals, such as methyl ε-caprolactone. Desirably, the caprolactone polyester has a molecular weight in the range of about 800 to about 2200, preferably about 1200 to about 2100, with corresponding hydroxyl numbers in the range of about 140 to about 45 and about 95 to about 55, respectively.

The adipates and azelates preferably have a molecular weight in the range of about 800 to about 2200 with a corresponding hydroxyl number in the range of about 140 to about 50.

Various suitable glycols for the preparation of the polyester polyols include straight chain aliphatic hydrocarbon diols, preferably hydroxyl-terminated diols, and alkylene ether glycols, preferably hydroxyl-terminated, for preparing the caprolactone polyesters, the adipates and azelates. Representative of the straight chain aliphatic hydrocarbon hydroxyl-terminated diols are 1,4-butane diol, 1,5-pentane diol, 1,6-hexane diol, 1,7-heptane diol. Representative of the alkylene ether glycols is diethylene glycol. The hydrocarbon diols are generally desired for the adipates and azelates with the 1,4-butane diol and 1,6-hexane diol being preferred. The caprolactone polyester of ε-caprolactone and diethylene glycol and polyesters selected from tetramethylene adipate, 1,6-hexane diol adipate, tetramethylene azelate and 1,6-hexane diol azelate are particularly desirable. The tetramethylene adipates and azelates are, of course, prepared from 1,4-butane diol and appropriate acid.

The polyesters are typically formed at a temperature of from about 50°C to about 300°C and preferably in the range of about 120° and 200°C. A catalyst can be used to increase the reaction rate, if desired. For a more detailed description of preparation of various suitable caprolactone polyesters, reference is made to U.S. Pat. No. 2,933,478.

The resilient polyureaurethane composition can be prepared by first reacting the polyether polyol or polyether polyol and polyester polyol with the diisocyanate under substantially anhydrous conditions at a temperature of from about 100° to about 130°C for about 30 to about 60 minutes. This reaction can be conducted at atmospheric or above or below atmospheric pressure. A catalyst can be added to the diisocyanate-polymeric polyol or polyol and polyester reaction mixture to reduce its reaction time. When such a catalyst is used, it is usually added to the reaction mixture before the addition of the diisocyanate or with the addition of the diisocyanate. Various catalysts can be used exemplary of which are the amine catalysts, such as triethyl amine, n-methyl morpholine and n-ethyl morpholine.

The diamine curative is then added to and mixed with the polymeric product of this reaction sometimes called a prepolymer, under essentially anhydrous conditions. The resulting polyureaurethane reaction mixture is then cast in a suitable mold and cured to form the shaped resilient polyureaurethane composition of this invention. The said reaction mixture can be cured at about 20° to about 50°C, although faster cures can be obtained at higher temperatures, for example, about 50° to about 200°C. Normally the reaction mixture is allowed to cure at 125°C from 16 to about 24 hours.

When the shaped resilient polyureaurethane composition is prepared by pouring the polyureaurethane reaction mixture into a mold having the desired configuration and then curing the polyureaurethane reaction mixture, metal plates suitable for use as force-receiving plates for the shock absorbing device of this invention can be placed in the mold before curing the polyureaurethane reaction mixture. If desired, a suitable bonding cement such as a phenolic or polyester-polyisocyanate adhesive may be applied to the metal plates. Exemplary are the cements taught to be useful in U.S. Pat. No. 2,992,939 and Australian Pat. No. 256,373. By curing the polyureaurethane reaction mixture in the presence of the said metal plates, a metal plate is adhered to at least one of the force-receiving surfaces of the polyureaurethane member to form a laminate of the structure shown in FIGS. 1 and 2, for instance. Such metal plates generally conform to the planar dimensions of the member's force-receiving surfaces and have a thickness of about 1/16 to about ½ inch, preferably about ⅛ to about ¼ inch, or about 100 to about 200 mils. Preferably, steel plates are used such as hot-rolled mild steel having a carbon content in the range of from about 10/15 to about 10/30 (Society of Automotive Engineers' (SAE) classification).

For further understanding of the invention, reference may be had to the accompanying drawing in which:

FIG. 1 is a perspective view illustrating one form of the shock absorbing device;

FIG. 2 is a vertical longitudinal sectional view further illustrating the shock absorbing device;

FIG. 3 is a vertical longitudinal sectional view of shock absorbing devices of the type shown in FIG. 1 and FIG. 2 placed in a supporting cylinder and disposed transversely to a compression force and compressed to about 40 percent of their original height.

Referring to the drawings, the shock absorbing devices or units shown in FIG. 1 and FIG. 2 comprise the

improved resilient cured polyureaurethane member 1 adhered to two opposite and substantially parallel force-receiving hot-rolled mild steel plates 2 and 3. If desired, a hole having a diameter of about 0.7 to about 1.3 inches extending from one force-receiving surface to the other can be provided through the member for mounting purposes. A portion of the side-wall of the resilient polyureaurethane member is concave in the form of an indentation such as a V-shaped groove 4. The ratio of the volume displaced by the groove to the volumes of the polyureaurethane member plus that displaced by the groove times 100 is about equal to the percent compression anticipated. A suitable rail-road draft gear can be formed as illustrated in FIG. 3 under a suitable compression load where the resilient cured polyureaurethane members are deformed and their sidewalls forced laterally outward 5.

The practice of this invention is further illustrated by reference to the following example which is intended to be representative rather than restrictive of the scope of the invention. Unless otherwise indicated, all parts and percentages are by weight.

## EXAMPLE I

Experiments A–B were conducted by first charging to reactors A–B respectively, under essentially anhydrous conditions, various amounts of polyether and polyester polyols consisting of polytetramethylene ether glycols having molecular weights of about 1000 and of about 2000, and polyester of ε-caprolactone and diethylene glycol having a molecular weight of about 2000. The polyester polyols had acid numbers of less than about 0.5. The mixtures of polyols were stirred (as a precautionary measure to remove any potential moisture) under reduced pressure at about 110°C for about 1 hour. To the polyol mixtures were then added various amounts of 1,5-naphthalene diisocyanate and the mixture stirred and allowed to react under reduced pressure at about 110°C for about 45 minutes. To the mixture was then added various amounts of 2,2'-diaminodiphenyldisulfide, (molten), [bis(2-AMP)DIS] or (2-methylpropyl)-4-chloro-3,5-diaminobenzoate. The following Table 1 illustrates the mixture make-up.

TABLE 1

| | A | B |
|---|---|---|
| Polytetramethylene ether glycol (1000) | 20 | 25 |
| Polytetramethylene ether glycol (2000) | 40 | 75 |
| ε-caprolactone polyester (2000) | 40 | — |
| 1,5-naphthalene diisocyanate | 23.8 | 25.4 |
| Bis(2-AMP)DIS | 11.3 | — |
| (2-methylpropyl)-4-chloro-3,5-diaminobenzoate | — | 12.7 |

The reaction mixtures were then immediately poured or cast into molds in which had been inserted two circular hot-rolled steel metal plates having an SAE classification of about 10/20 and having diameters of about 6.5 inches and thicknesses of 135 mils. The plates had been coated with a polyester-polyisocyanate type of adhesive to enhance their adhesion to the cast polyureaurethane. The mixtures were cured in the molds at about 115°C for about 22 hours to provide polyureaurethane steel laminates as shock absorbing units similar to that shown in FIGS. 1–3 in the drawing of this specification, the shaped resilient polyureaurethane compositions having diameters of 6.5 inches and thicknesses of 1.5 inches. Their sidewalls were in the shape of a V-shaped groove having a volume equal to about ⅜ of the polyureaurethane.



The shock absorbing units deflected (compressed) about 0.44 to about 0.55 inch at about 25°C upon the application of about 1800 pounds per square inch uniformly to the surface areas of steel force-receiving plates. Actually, the test was conducted by placing two of such units in series under test, and their total deflection was about 0.88 to about 1.1 inch.

Shock absorbers having polyureaurethane members prepared according to this invention but having too low a mole ratio of diisocyanate to polymeric polyester typically deflect more than about 0.6 inch when subjected to this test and are therefore generally considered too soft. Such shock absorbing units when used in a railroad car draft gear typically absorb insufficient amounts of energy and, thus, are usually fully compressed before sufficient compressive force energy is absorbed by the gear during usage. Such shock absorbers having a polyureaurethane member having too high a ratio of diisocyanate to polymeric polyester typically deflect less than about 0.30 inch when subjected to this test and are therefore generally considered too hard. When used in a railroad draft gear, they typically absorb an insufficient amount of energy before transmitting the energy, or force resulting from coupling the railroad car, through the draft gear and also break down early during usage.

Portions of each of the polyureaurethane reaction mixtures were cured and shaped to form discs having diameters of about 1.13 inch and thicknesses of about 0.5 inch. At about −40°C a pressure applied to their flat surfaces of about 5800 to about 7300 pounds per square inch was required to compress the individual discs about 40 percent of their original thicknesses. At 24°C such a compression required from about 2000 to about 2500 pounds per square inch with a maximum of about 2700 being desired.

Thus, the shock absorbing units had desirable load deflections or compressions for use in railroad draft gears for a wide range of temperatures such as from about −20°C to about 25°C and preferably up to about 50°C.

The shock absorbing units are further desirably characterized by a −35°C hammer drop test and by an AAR endurance test.

A hammer drop test is described by first vertically loading a draft gear cylinder with 10 of the shock absorber units or pads similar to FIGS. 1–3 in series to a shock load with their adherent metal force-receiving plates facing each other to form a draft gear. A 27,000 pound hammer is dropped onto the end of the vertically positioned gear from several heights. The impact shock is measured, typically expressed as the height the hammer is dropped in inches, and the capacity of the gear is determined. The capacity of the gear is measured at the point where the gear "bottoms out", i.e., when it starts to transmit shock directly from the hammer drop rather than cushion and absorb the shock force. Thus, the gear can typically "bottom out" with a 27,000 pound hammer being dropped from a height of about 18 inches for a 40,000 foot pound shock force. The draft gear is then cooled to −35°C and drop hammered three times with the 27,000 pound hammer for the −35°F hammer drop test. The capacity is measured and the gear disassembled followed by examining the pads. A criteria for failing the hammer drop test is

deterioration of the pads such as cracking, particularly at −35°C, or by bottoming out at a shock load less than about 40,000 foot pounds at about 25°C.

The AAR Endurance Test (American Association of Railroads) can be referred to as AAR Spec. M-901-E Endurance Test. The test is generally similar to the −35°C hammer drop test but starting at room temperature or about 25°C. A 27,000 pound hammer is dropped at variable vertical heights of from about 1 to about 30 inches over a period of time until 35 million foot pounds of energy have been expended upon the gear which typically comprises 10 of the shock absorbing units or pads. The capacity of the gear is measured both at the beginning and at the end of the test as well as periodically during the test. The gear capacities before, during and after the test are then compared to determine any changes in capacity which the gear may undergo. The gear is then disassembled and inspected for deterioration of the pads. Appreciable loss of capacity or deterioration of the pads, such as by excessive cracking, are criteria for failing the endurance test. It is preferred that the gear, when composed of 10 of the pads, has a capacity of at least about 40,000 foot pounds before bottoming out, or a capacity of about 4,000 foot pounds per pad at about 25°C. In this test, the 27,000 pound hammer shocks are applied gradually over a period of time to prevent excessive heat build-up because the gear heats up considerably after each hammer drop.

Shock absorbing units prepared from experiments A and B successfully passed the compressive cycling test for at least about 500 cycles where a unit was alternately compressed under constant compression alternating between about 8 to about 12 percent down to about 45 to about 55 percent of its original polyureaurethane height. More specifically, a unit is compressed from about 8 to about 12 percent down to said 45 to 55 percent and allowed to return to its 8 to 10 percent compression. This may take about 30 seconds. About 4½ minutes later the cycle is repeated. Thus, a cycle takes about 5 minutes. Typically a larger force is required to compress the unit during its first cycle with such force diminishing somewhat for the next 10 to 50 cycles. Then the force tends to level out or stay somewhat constant until the unit begins to break down. At this point, the required force typically rather quickly reduces over a span of a relatively few cycles. Therefor, a typical measure of cycles which a unit suitable for a railroad draft gear can withstand or endure is the number of such five-minute cycles it can withstand until the maximum compressive force necessary to compress it from a compression of about 8 to about 12 percent of



its original polyureaurethane uncompressed height down to about 45 to about 55 percent of such height has reduced or dropped about 20 percent of the force required for the first compression cycle.

The shock absorbing units of this invention and particularly as described in this example, have unique utility as railroad draft gear shock absorbing units. In practice, the shock absorbing device of a railroad draft gear is typically assembled by the series loading of 8 to 14, preferably 10, of the shock absorbing units of this invention and preferably of the prescribed 6.5 inch diameter shaped polyureaurethane disc having the 1.5 inch thickness and V-grooved sides, followed by placing the units in the device under about 20,000 pounds force for operational use in the railroad car.

In the practice of this invention, it is typically desired that the polyureaurethane contain an antioxidant amount of an antioxidant. Thus, it may be typically desired that the polyureaurethane contain in the range of about 0.5 to about 3 and more preferably about 1 to about 2 weight percent of an antioxidant such as an amine or a hindered phenolic type. Usually an amine antioxidant is satisfactory. Usually the antioxidant is mixed with a diol diisocyanate mixture or product or more preferably is simply mixed with the polyol. The addition of the antioxidant is primarily to enhance the maintenance of the shock absorber's desired properties over a long period of time.

## EXAMPLE II

Experiments C–G are conducted by first charging to reactors C–G respectively, under essentially anhydrous conditions, various amounts of polyether and polyester polyols consisting of polytetramethylene ether glycols having molecular weights of about 1000 to about 2000, a polyester of ε-caprolactone and diethylene glycol having a molecular weight of about 2000 and a polytetramethylene azelate having a molecular weight of about 2000. The mixtures of polyols are stirred (as a precautionary measure to remove potential moisture) under reduced pressure at about 110°C for about an hour. To the polyol mixtures is then added various amounts of 3,3'-dimethyldiphenylenemethane-4,4'-diisocyanate, 3,3'-bitolyene-4,4'-diisocyanate and 1,5-naphthalene diisocyanate. The mixtures are stirred and allowed to react under reduced pressure at about 110°C for about 45 minutes. To the mixtures are then added various amounts of (2-methylpropyl)-4-chloro-3,5-diaminobenzoate or 2,5-dichloro-1,3-phenylene diamine. The following Table 2 illustrates the mixture make up showing the amounts in parts by weight of the various materials for experiments C–G.

TABLE 2

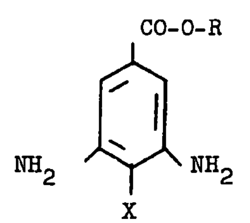
| | C | D | E | F | G |
|---|---|---|---|---|---|
| Polytetramethylene ether glycol (1000) | 30 | 25 | 30 | 25 | 25 |
| Polytetramethylene ether glycol (2000) | 35 | 30 | 40 | 35 | 40 |
| Polytetramethylene azelate (2000) | 35 | 45 | 30 | — | — |
| Polycaprolactone ester (2000) | — | — | — | 40 | 35 |
| 3,3'-Dimethyldiphenylenemethane-4,4'-diisocyanate | 35.2 | — | 35.2 | — | — |
| 3,3'-bitolyene-4,4'-diisocyanate | — | 32.2 | — | 31.4 | — |
| 1,5-naphthalene diisocyanate | — | — | — | — | 25.0 |
| (2-methylpropyl)-4-chloro-3,5-diaminobenzoate (molten) | 12.9 | 12.4 | — | — | — |
| 2,5-dichloro-1,3-phenylene diamine | — | — | 9.38 | 8.60 | 8.59 |

The reaction mixtures are then immediately poured or cast into molds in which are inserted two circular hot-rolled steel metal plates having an SAE classification of about 10/20 and having diameters of about 6.5 inches and thicknesses of 135 mils. The plates are coated with a polyester-polyisocyanate type of adhesive to enhance their adhesion to the cast polyureaurethane. The mixtures are cured in the molds at about 115°C for about 22 hours to provide polyureaurethane steel laminates as shock absorbing units similar to that shown in FIGS. 1–3 in the drawing of this specification, the shaped resilient polyureaurethane compositions having diameters of 6.5 inches and thicknesses of 1.5 inches. Their sidewalls are in the shape of a V-shaped groove having a volume equal to about ⅗ of the polyureaurethane.

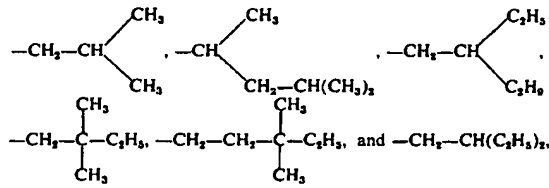
While certain representative embodiments and details have been shown for the purpose of illustrating the invention, it will be apparent to those having skill in this art that certain changes and modifications may be made therein without departing from the spirit or scope of the invention.

What is claimed is:

1. Shock absorbing unit which comprises a shaped resilient polyureaurethane composition characterized by (A) withstanding compressive cycling for at least about 500 cycles under constant compression alternating between a maximum of about 45 to about 55 percent and a minimum of about 8 to about 12 percent of its original uncompressed height, (B) requiring at least about 4200 pounds per square inch to compress said unit 55 percent of its original uncompressed height after 500 cycles of said compressive cycling, and (C) deflecting from about 0.3 to about 0.6 inch, preferably from about 0.4 to about 0.5 inch, at about 25°C upon the application of about 1800 pounds per square inch uniformly to the end surface areas of the said resilient compositions in its uncompressed state when the said composition is a generally disc-shaped cylindrical element with circular parallel end surfaces, said surfaces covered and adhered to circular steel plates, having a diameter of about 1.5 inches and a sidewall connecting the end surfaces substantially in the form of a V-shaped groove having substantially equal length sides, the said groove extending between the said end surfaces, the volume of said element being between 150 percent of the volume of the said groove, where said resilient shock absorbing polyureaurethane composition is prepared by the method which comprises reacting a diamine selected from 2,2'-diaminodiphenyldisulfide, 2,5-dichloro-1,3-phenylene diamine and a substituted diamine of the formula




where R is selected from alkyl radicals selected from methyl, ethyl, n-propyl, n-butyl, isopropyl, isobutyl, n-pentyl, isopentyl, hexyl, heptyl, octyl and decyl radicals, branched chain radicals selected from



aryl radicals selected from tolyl, phenyl and halophenyl radicals, aralkyl radicals selected from benzyl and α-alkyl benzyl radicals, where the alkyl group has 1 to 6 carbon atoms, and cycloalkyl radicals selected from cyclopentyl, cyclohexyl, 4-t-butylcyclohexyl and cycloheptyl radicals, and X is a chlorine radical, with the reaction product of (A) a diisocyanate selected from 1,5-naphthalene diisocyanate when the diamine is selected from 2,2'-diaminodiphenyldisulfide or a diisocyanate selected from 1,5-naphthalene diisocyanate, 3,3'-bitolylene-4,4'-diisocyanate and 3,3'-dimethyldiphenylenemethane-4,4'-diisocyanate when the diamine is selected from said substituted diamine and 2,5-dichloro-1,3-phenylene diamine, and (B) a mixture of at least one polymeric polyol with a total average molecular weight of about 1500 to about 2100 which comprises (1) polyols having a molecular weight of about 1800 to about 2200 selected from (a) about 65 to about 100 weight percent of a polyether polyol or (b) about 65 to about 100 weight percent of a polyol mixture comprising (i) about 35 to about 65 weight percent polyether polyol and (ii) about 65 to about 35 weight percent polyester polyol or (c) about 65 to about 95 weight percent of said polyether polyol-polyester polyol mixture and correspondingly (2) about 35 to about zero or 5 weight percent of at least one of a polyether polyol and polyester polyol having a molecular weight of about 800 to about 1250, wherein said polyether polyol is selected from polytetramethylene ether glycol and polypropylene ether glycol, and said polyester polyol is selected from (i) caprolactone polyesters prepared from caprolactones containing 6 to 8 carbon atoms and glycols containing 4 to 7 carbon atoms and (ii) azelates of azelaic acid and glycols containing 4 to 7 carbon atoms, where the ratio of isocyanato groups to the sum of hydroxyl groups of the polyols is about 1.7 to about 2.5, where the ratio of primary amino groups of the diamine to excess isocyanato groups over the sum of said hydroxyl groups is about 0.6 to about 1.1 and where the acid number of the polyols is less than about 1.

2. The shock absorbing unit of claim 1 where said substituted diamine is selected from (methyl)-4-bromo-3,5-diaminobenzoate, (methyl)-4-iodo-3,5-diaminobenzoate, (ethyl)-4-fluoro-3,5-diaminobenzoate and (2-methylpropyl)-4-chloro-3,5-diaminobenzoate.

3. The shock absorbing unit of claim 1 where substituted diamine is (2-methylpropyl)-4-chloro-3,5-diaminobenzoate.

4. The shock absorbing unit of claim 1 where said diamine selected from 2,2'-diaminodiphenyldisulfide and said substituted diamine is reacted with the reaction product of 1,5-naphthalene diisocyanate and a polyether polyol-polyester polyol mixture selected from (a) polytetramethylene ether glycols having an average molecular weight of about 900 to about 1100 or a mixture with molecular weights of about 900 to

about 1100 and of about 1900 to about 2100 and (b) at least one of the polyester polyols selected from hexane diol adipate and azelate having molecular weights of about 800 to about 1200, tetramethylene azelate having a molecular weight of about 1800 to about 2200 and polyesters of $\epsilon$-caprolactone and diethylene glycol having molecular weights of about 1100 to about 1400 and about 1800 to about 2200.

**5.** The shock absorbing unit of claim **4** where said substituted diamine is (2-methylpropyl)-4-chloro-3,5-diaminobenzoate.

**6.** The shock absorbing unit of claim **1** where said diamine selected from 2,5-dichloro-1,3-phenylene diamine and said substituted diamine is reacted with the reaction product of a diisocyanate selected from 1,5-naphthalene diisocyanate, 3,3'-bitolylene-4,4'-diisocyanate and 3,3'-dimethyldiphenylenemethane-4,4'-diisocyanate and a polyether polyol-polyester polyol mixture selected from (a) polytetramethylene ether glycols having an average molecular weight of about 900 to about 1100 and of about 1900 to about 2100 and (b) at least one of the polyester polyols selected from hexane diol adipate and azelate having molecular weights of about 800 to about 1200, tetramethylene azelate having a molecular weight of about 1800 to about 2200 and polyesters of $\epsilon$-caprolactone and diethylene glycol having molecular weights of about 1100 to about 1400 and about 1800 to about 2200.

**7.** The shock absorbing unit of claim **6** where said substituted diamine is (2-methylpropyl)-4-chloro-3,5-diaminobenzoate.

**8.** The shock absorbing unit of claim **1** having a generally disc-shaped cylindrical element with circular parallel end surfaces having diameters of about 6.5 inches, a height of about 1.5 inch, and a side-wall connecting the end surfaces substantially in the form of a V-shaped groove having substantially equal length sides, the said groove extending between the said end surfaces, the volume of the solid portion of the said element being about 150 percent of the volume of the said groove, where the said adherent rigid force-receiving plates are steel plates, and where said compressive cycling is composed of cycles which comprise alternating between a maximum of about 45 to about 55 percent and a minimum of about 8 to about 12 percent of its original uncompressed height over a period of about 30 seconds for a complete cycle followed by a pause between cycles.



**9.** The shock absorbing unit of claim **8** where said polyether polyol-polyester polyol mixture is selected from (a) polytetramethylene ether glycols having an average molecular weight of about 900 to about 1100 or a mixture with molecular weights of about 900 to about 1100 and of about 1900 to about 2100 and (b) at least one of the polyester polyols selected from hexane diol adipate and azelate having molecular weights of about 800 to about 1200, tetramethylene azelate having a molecular weight of about 1800 to about 2200 and polyesters of $\epsilon$-caprolactone and diethylene glycol having molecular weights of about 1100 to about 1400 and about 1800 to about 2200.

**10.** The shock absorbing unit of claim **9** where said substituted diamine is (2-methylpropyl)-4-chloro-3,5-diaminobenzoate.

**11.** The shock absorbing unit of claim **9** connected with a plurality of such units to form a railroad draft gear containing 8 to 12 of such units loaded in a cylinder in series to a shock load with their rigid force-receiving plates facing each other.

**12.** A method of preparing the shock absorbing unit of claim **9** which comprises reacting the polyureaurethane reactants in a mold having the required configuration and having metal force-receiving plates inserted therein.

* * * * *